IPSC

BURNER MODIFICATION

PROGRAM REPORT

for

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Ipscblup.rep

Ideas in progress

RJM

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INTRODUCTION

The IPSC boilers have been analyzed for design improvements to upgrade the durability of the coal fired burners. The existing burners experience significant mechanical stress. The inner and outer air registers and associated parts have severe thermal distortion which causes the registers to become inoperable. The aerodynamics have been examined and a swirler design made for the inner air path. Revised register settings are recommended along with an outer band to restrict inlet flow area into the outer register. The upgrade burner will have lower swirl generation in both registers. This will inhibit the tendency for gas recirculation that is evident in the existing design. The secondary air velocity with the upgrade burner aerodynamics will shape the recirculation zone to reduce NO_X formation. Structural modifications are recommended to the burner to reduce thermal stress. These incorporate a segmented outer register back plate along with radial and axial positioning means to hold the slip fit design components. Materials and thickness will remain the same as used in the existing design.

BURNER DESCRIPTION

The IPSC coal fired unit is a B&W design with 48 burners rated at 100% load for 6,100,000 lbm/hr steam leaving the superheater at 2515 psig and 1005°F. The existing burner design is shown in Figures 1 and 2 based on drawing 294361E-12. The B&W proposed design is shown on Figures 3 and 4 based on drawing SK41791E-O. A summary of the B&W proposed modification is listed in Figures 5 and 6. The recommended burner design modifications including the swirler are shown in Figure 7.

PROBLEM DESCRIPTION

The existing design problem areas are listed in Figure 8. A complete set of photos showing burner distress is in Appendix I.

The warping is caused by extreme thermal stress. Subsequent jamming of the register vanes and air slide occurs as well as relative movement which damages the rope packing air seal.

It can be seen in the photo that the inner sleeve appears distorted in the arc immediately downstream of the ignitor. It should be checked to see if any of the ignitors are firing when retracted in the burner instead of forward of the coal pipe end.

The coal pipe photos show that severe melting, resulting in egg shape ends of the pipe, is associated with coal pipe fires. In fact a photo shows that severe heat in the coal pipe elbow at the burner cover plate occurred during one fire.

AERODYNAMIC ANALYSIS

The existing burner operating conditions and register settings are shown in Figure 9. The inner air sleeve is set open 3 to 5 inches in the existing burners. Primary airflow passes with the pulverized coal. The burner secondary airflows through the outer and inner registers. Evaluation was made of the swirl number for the secondary airflow. Definition of the swirl number and its importance is listed on Figure 10. For optimum combustion and NO_X reduction, a value of 0.8 to 1.0 is desirable for the inner and outer flow. Lower outer flow swirl (.5 or less) may cause potential coal ignition problems. Higher values (approx. 1.5 or more) create over swirl, which results in an improperly sized recirculation zone and the potential of gas recirculation into the air sleeve and throat zones. The recirculation parameter is defined in Figure 11 and is a measure of the potential for gas recirculation into the air ducts. The baseline analysis for the existing burner and settings shows the potential for hot gas ingestion, which can aggravate the thermal loading on the register parts. The recommended solution was to reduce swirl in both paths by changing register settings and using a properly designed fixed vane swirler in the inner path.

Computer output of the aerodynamic calculations is listed in Appendix II for the baseline and swirler analysis.

Baseline Analysis

The secondary airflow through the registers was evaluated at a windbox-to-furnace pressure differential of 2.0 inches of water and air inlet temperature of 650°F. The airflow is a function of register or spin vane setting angle. The outer register airflow, shown in Figure 12, is 19.1 lbm/sec at 65 degrees vane exit flow angle. The inner register airflow, shown in Figure 13, is 15.2 lbm/sec at 60 degrees vane exit flow angle, corresponding to the outer register setting. The primary airflow in the coal pipe is 10.4 lbm/sec. Total flow per burner is 44.7 lbm/sec and for the furnace with 42 burners in service is 6,760,000 lbm/hr.

The swirl number is 1.64 for the outer register, shown in Figure 14, at 65 degrees vane exit flow angle. The swirl number is 1.36 for the inner register, shown in Figure 15, at 60 degrees vane exit flow angle. Swirl number for both flows is excessive based on our experience, which will produce a combustion internal recirculation that is too large. Also, since the recirculation parameter is calculated to have a negative value, there is the potential for hot gas ingestion in the duct exit The recirculation parameter is -0.2 inch of water, shown in Figure 16, for the outer duct at 65 degrees vane exit flow angle. The recirculation parameter is -0.4 inch of water, shown in Figure 17, for the inner duct at 60 degrees vane exit flow angle.

Swirler Analysis

The approach for the aerodynamic improvement was to design a swirler to provide approximately 1/3 of the burner airflow at a swirl number of 0.85.

A swirler design was employed which uses curved vanes with axial inlet angle and exit angle variation from 40° at the hub to 65° at the tip. Mechanical design of the swirler is discussed in a later section.

The swirler is used for the inner flow, which strongly effects the shape of the combustion recirculation zone. The spin vanes of the inner register will be set full open (i.e. 0 degrees flow angle) to provide axial flow into the swirler. The outer register will be set at a lower vane exit flow angle to provide reduced swirl number.

With reduced swirl in both secondary flows, the required windbox-to-furnace differential pressure is lower than for the existing burner design and register settings. Two cases were evaluated with the swirler to obtain the same secondary airflow rates as in the existing design.

a) Pressure differential of 1.19 inch of water.

In this case the outer register vane exit flow angle was set at 56 degrees. Outer register airflow was calculated to be 19.0 lbm/sec which produces a higher than desirable swirl number of 1.13 and a positive recirculation parameter of 0.25 inch of water. The flow in the inner register with the air slide open 10 inches was calculated to be 15.5 lbm/sec. The swirler generates the design swirl number of 0.85. The recirculation parameter is positive and equal to 0.98 inches of water at the swirler hub and 1.80 inches of water at the coal pipe O.D. This case avoids negative recirculation parameter, but has a higher swirl number than necessary for the outer register airflow.

b) Pressure differential of 1.99 inch of water.

In this case the pressure differential is increased, which will help provide better burner to burner airflow uniformity. This is accomplished by putting a band around the inlet to the outer register to block approximately 50 percent of the flow area as calculated at the register air door restriction. The inner register air slide will be set less open than in case (a). The band permits setting the outer register vane exit flow angle at 50 degrees. Outer register airflow was calculated to be 1.91 lbm/sec, with a swirl number of .91 and a positive recirculation parameter of .40 inch of water. The flow in the inner register with the air slide open 5 inches was calculated to be 15.0 lbm/sec. The swirler generates the design swirl number of 0.85. The recirculation parameter is positive and equal to 1.05 inches of water at the swirler hub and 1.82 inches of water at the coal pipe O.D. This case also avoids negative recirculation parameter, but has a desirably lower value of swirl number in the outer duct. The pressure differential of 1.99 inches of water is equal to the existing baseline operating value. This case (b) is recommended and will require addition of a band on the outer air register inlet, which will restrict the open flow area to 1300 square inches (5.9" open slot width).

BURNER STRUCTURAL ANALYSIS

Test Data

The existing design of Unit 2 burners was modified and set as listed in Figure 18. Measured temperatures at full load with burners "in" and "out" of service are shown in Figure 19.

Thermal Design Conditions

The heat transfer analysis design conditions are summarized in Figure 20. In service, a maximum air side convective heat transfer coefficient (HTC) of 12 Btu/hr ft^2°F was used on the duct walls, based on velocity corresponding to 100% load airflow. Reduced (HTC) values were used in the lower velocity regions and on the surfaces exposed to the windbox. Radiation load was based on a 3200°F flame temperature and shape factors to the surfaces from mid flame position. The In-Service condition matched the measured temperature on the back plate. The Out-Of-Service conditions predicted temperatures higher than measured, which were intended to predict worst case structural loads.

Analytical Model

A finite element heat transfer and stress analysis was performed on the existing burner design, (294361E-12), the B&W proposed burner design, (SK 41791E-0), and on our recommended modified design using the COSMOS/M finite element computer program.

For the existing design, the inner sleeve, back plate, throat sleeve and front plate were modeled as an assembly using 392 axisymetric ring elements. Nodal displacements linking the front plate to the back plate were used to simulate the outer register assembly. Radiant heat flux and cooling airflows were simulated using element heat generation and surface convection coefficients. See Figure 21.

For the proposed design, only the slip-fit back plate was analyzed (using 84 axisymetric ring elements), since the unrestrained inner sleeve will be free to grow thermally and is therefore essentially stress-free. Similarly the slip-fit feature of the front plate/throat sleeve attachment relieves the thermal stress build-up between these parts so that further analysis of these parts is not required.

For the recommended modified design only the segmented back plate panels were analyzed using a 90° circular plate model consisting of 228 plane stress elements 1/2 inch thick. See Figure 22.

For the existing design and the B&W proposed design, the In-Service operating condition was evaluated based on the prescribed thermal design heat flux and airflow rates. Out-Of-Service operation was based on the full In-Service heat flux, but with the reduced (shut-down) airflow rates.

For the recommended segmented back plate design, the In-Service operating condition was evaluated based on the temperature field calculated for the proposed design. The Out-Of-Service operating condition was based on the calculated temperature field adjusted to match the maximum temperature measured for this condition.

For all cases, elastic analyses were performed. Material properties used to evaluate the designs are listed in Figures 23 through 26.

<u>Existing Design: In-Service</u>

The results of the heat transfer analysis are shown in Figure 27. It is seen that temperatures along the inner sleeve are generally low, but indicate a local hot spot of 875°F caused by heat flow from the back plate. A maximum temperature of 1060°F is predicted on the back plate just outboard of the inner sleeve, which is within the range 850-1220°F, measured for this condition. Low temperatures were also predicted in the throat sleeve and forward plate.

The resulting thermal growth is indicated in Figure 27A which shows the deformed finite element model, scaled for clarity, superimposed over the undeformed (room temperature) model. The radial growth of the inner sleeve is slightly greater at the back plate attachment (0.160 inch) than at either end due to the locally higher temperature at that location. Similarly, radial growth of the back plate outer diameter is somewhat larger than that of the front plate (0.301 vs 0.245 inch), skewing the outer register connecting bars (not shown). Finally it is seen that the forward end of the throat sleeve tends to move axially (0.168 inch) as well as radially (0.221 inch), assuming no restraint from the air slip seals (not shown).

The results of the stress analysis in Figure 28 showed generally low stresses along the inner sleeve except for a high (22,000 psi) local stress concentration due to the thermal growth of the back plate. The high compressive tangential stress shown in the back plate results from the hot material near the inner radius expanding against the restraint of the colder material near the outer register. For a thin plate, this type stress field will result in a circumferential coning or buckling distortion of the plate. Fracture of the weld is expected at this stress level and temperature (see photos in Appendix I) with subsequent separation from the inner sleeve and jamming of the outer register vanes.

The stresses shown in the front plate and throat sleeve are generally low except for a moderate local stress concentration at the attachment weld. The peak stress of 7600 psi at 740°F is well within the allowable stress limits previously given in Figure 23.

The results of this analysis are summarized in Figure 29.

Existing Design: Out-Of-Service

For the Out-Of-Service analysis the high radiant heat flux of the In-Service condition was imposed in order to simulate a "worst-case" loading condition.

The heat transfer analysis shows higher temperatures throughout the design, predicting a peak temperature of 1760°F on the back plate, which far exceeds the measured temperature range of 980°F - 1285°F typical for this condition. Figure 30.

The thermal growth pattern is similar to the In-Service case, but with proportionately greater deformation values.

Based on this "worst-case" temperature field, the stress analysis, Figure 31, shows very much higher compressive tangential stresses in the back plate which would severely aggravate the circumferential coning/buckling distortion already predicted to occur during In-Service operation. Additional, and more certain, separation from the inner sleeve would, therefore, be predicted with aggravated distortion of the plate and jamming of the outer register vanes.

Fracture of the front plate/throat sleeve attachment would also be predicted.

The results of this analysis are summarized in Figure 32.

<u>B&W Proposed Design: In-Service</u>

For this design, a finite element analysis was performed only for the slip-fit back plate since the inner sleeve is now free to grow thermally resulting in an essentially stress-free state. A similar argument applies to the separated front plate and throat sleeve where low temperatures and stresses are now expected.

The heat transfer results, Figure 33, shows the maximum temperature on the back plate is now at the inner radius, and is 25°F hotter than the existing design due to the gap between it and the inner sleeve.

The maximum compressive stress, Figure 34, has also increased slightly resulting in a slightly stronger tangential stress field than in the existing design, and therefore the same coning/buckling behavior persists in the proposed design. Jamming of the outer register vanes is also expected.

Because coning/buckling distortion is the predicted failure mode, upgrading the material to one of higher strength will have negligible beneficial effect. Increasing the thickness from 1/2 inch to 5/8 inch will provide some, but very minor, additional resistance to buckling.

The results of this analysis are summarized in Figure 35.

Proposed Design: Out-Of-Service

This analysis gave generally similar results to the In-Service results, but again with higher temperatures and stresses, further aggravating the predicted coning/buckling distortion of the plate. See Figures 36 and 37.

The results of this analysis are summarized in Figure 38.

Recommended (RJM) Modified Design: In-Service

RJM's recommended back plate design (see Figure 39) consists of four, 90° segmented panels, slip-fitted to each other and to the inner sleeve and outer register assembly of the burner assembly, previously shown in Figure 7. Overlapping plates are shown installed between the segments to minimize airflow through the gaps. Radial support bars, two for each panel, position the panels and prevent binding of the panels during operation.

A finite element stress analysis of the four panel design was performed based on the calculated In-Service temperature field. The resulting thermal growth of these panels is shown in Figure 40. It is seen that each panel is now free to grow, and for this loading, expands 0.290 inches circumferentially at the inner radius, thereby eliminating the circumferential coning/buckling failure mode of the full plate design.

The stress results of Figure 41, shows that the edges are stress free, and that the material along the inner radius is now in tension, and effectively balanced by an adjacent compressive field. Similar tensile/compressive stress fields occur in the outer region of the plate.

Note that the calculated (elastic) stresses cannot, in practice, exceed the material's yield strength, so that the peak tensile stress is limited to 18,300 psi at this temperature. See Figure 42. Plastic straining is therefore very minor, and limited to a small region along the inner radius. Furthermore, the fixed thermal strains distributed within the panel will relax due to creep, causing a gradual reduction of the stresses throughout the panel. Residual stresses resulting from the plastic strains will also gradually disappear. Failure by low cycle fatigue or creep rupture is therefore not expected.

Recommended (RJM) Modified Design: Out-Of-Service

For the Out-Of-Service condition, the calculated temperature field was adjusted to match the maximum temperature typically measured for this condition, (1285F), plus the additional 25F predicted for the slip-fit plate.

The calculated free tangential growth at the inner radius is now 0.350 inch.

The peak (elastic) stress, again at the inner radius, as shown in Figure 43, is now limited to the material's yield strength of 16,200 psi at this temperature, resulting in additional, but still minor plastic strains around the inner radius. As for the In-Service case, creep relaxation will gradually reduce these stresses throughout the panel and the residual stresses resulting from the plastic strains will also gradually disappear.

The results of this analysis and back plate design features are summarized in Figure 44.

Structural Analysis Summary

It is concluded that, during In-Service operations, the high radial temperature gradient on the back plate of the existing design, with its associated high compressive tangential stress field, results in circumferential coning/buckling of the plate with subsequent weld failures and separation from the inner sleeve. Jamming of the outer register vanes is therefore also expected. Out-Of-Service operation further aggravates the damage.

For the B&W proposed design, the slip-fit on the back plate does not relieve the high compressive tangential stress field in the plate, so that the same (or slightly more severe) coning/buckling distortion is predicted. The slip-fit feature, however, does relieve the stresses in the inner sleeve, an well as in the front plate and throat sleeve.

It was also concluded that little, if anything, would be gained by upgrading the materials or in increasing the thicknesses of the proposed design.

It is recommended that the proposed slip-fit back plate be modified by segmenting the plate into 4 separate 90° panels so as to relieve the high tangential stress field and the associated circumferential coning/buckling distortion. Jamming of the outer register doors will then also be eliminated. To ensure free thermal growth of all four panels when installed, a tangential gap of 0.75 inch is recommended between panels, with radial gaps of 0.25 and 0.50 inch at the inner and outer radius respectively.

It is also recommended that the B&W proposed inner sleeve and slip-fit front plate/throat sleeve designs remain as proposed except for the inclusion of radial support bars.

The structural conclusion and recommendations are summarized in Figure 45.

Finally it is recommended that the same materials and thicknesses of the existing design be used through out. See Figure 46.

Swirl Nozzle Mechanical Design

The recommended Swirl Nozzle design consists of 40 vanes welded to an outer and inner shroud as shown in Figure 47. Both the outer and inner shrouds are segmented to permit free thermal growth of the assembly, thereby relieving the unit from high locked-in thermal stresses. A finite element analysis was therefore not performed since the structure is essentially free of thermal stresses.

Installation of the swirler is shown in Figure 48. Cutouts (dimensions and locations supplied by IPSC) in the swirler vanes permit clearance for the ignitor, scanner and observation port. These are mirror images for CW and CCW swirlers. The unit is attached at the outer shroud to the coal nozzle by 16 support straps, 2 per segment, to allow for free thermal growth between the (hotter) outer shroud and the (cooler) coal nozzle. Locking pins, fitted to the inner shroud, permit free radial and tangential thermal growth of the segments while constraining axial movement of the segments along the inner shroud. The swirler is constructed of high temperature stainless steel (SS310).

A summary of the swirler mechanical design features is listed in Figure 49.

CONCLUSIONS AND RECOMMENDATIONS

- Aerodynamic analysis indicates that high swirl numbers associated with the inner and outer register airflow are excessive (approximately 1.5) and have the potential for hot gas recirculation into the air cavities.

- A swirler for the inner airflow will lower and provide a constant swirl number of 0.85.

- A band (or perforated screen) is recommended for the outer register inlet to permit it to be operated at a reduced exit vane flow angle and thus reduce its swirl number.

- Structural analysis confirms that the high radial temperature gradient on the outer register back plate of the existing design with its associated high compressive tangential stress field results in circumferential coning/buckling of the plate with subsequent weld failures and separation from the inner sleeve. Jamming of the outer register vanes is therefore also expected.

- Recommended additional burner modifications are:

  - Segmented back plate panels with slip fit clips and radial positioning support.

  - Outer register vane assembly radial and axial position struts.

  - Throat sleeve with slip fit clips to register and radial position bars.

  - The separate outer ring design for the air seal will avoid radial thermal growth problems, but the air seal stop must be modified to permit axial movement.

- The same materials and thickness of the existing design can be used with the recommended burner design modifications.

- The recommended burner design modifications and material selections are being provided by the RJM Corporation, with the understanding that the final design decision and implementation shall be the responsibility of the Intermountain Power Service Corporation and the Babcock and Wilcox Company.

NO_X PREDICTION

As a means to estimate the impact of the MZ Flame Stabilizer and recommended register settings on NO_X emissions, RJM reviewed the latest relevant literature. In that review, several items became evident to us. For dual register burners, a significant amount of swirl number optimization work has taken place in recent years. Work done by LaRue in the early 80's showing outer swirl numbers of .6 to .7 and inner swirl numbers of 1.3 to 1.4 as being recommended practice or normal operation (Figures 50 and 51). Jeremiejczyk in 1978 shows studies over a range of both swirl numbers in the inner from 0 to .75 and the outer from .4 to 1.2. Jeremiejczyk and others also found that the outer swirl number should be greater than .5 to assure ignition point and flame stability with various coal types. Current operation at IPSC has the following swirl number settings:

Inner = 1.36

Outer = 1.64

The RJM recommendations for swirl numbers are:

Inner = .84 (MZ Stabilizer)
Outer = .91

The outer swirl number is larger than RJM's experience would dictate which is heavily influenced by prior investigators and their measured ignition/flame stability limits on oil and gas units. However, the fuel volatility and swirl number interdependence indicated by Jeremiejczyk clearly indicates higher swirl numbers would be beneficial for ignition and flame stability as the relative volatility of the fuel decreases. Based on this research, RJM has adjusted IPSC's swirl numbers accordingly.

In terms of NO_X, if we use Figure 52 with the closest comparisons to current IPSC swirl settings and RJM recommendations, a lower O_2 profile is seen to exist on the centerline with lower temperatures (perhaps 20°C). Ignoring the O_2 difference (which helps the MZ stabilizer NO_X), the temperature difference would indicate perhaps as much as 14% NO_X reduction using RJM scaling factors. Obviously, an exact NO_X prediction is not possible, but at least the recommended settings and hardware will tend to drive NO_X down from current levels; perhaps as much as the calculated 14%.

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INTERMOUNTAIN POWER PROJECT EXISTING DESIGN (REF. 294361E-12)

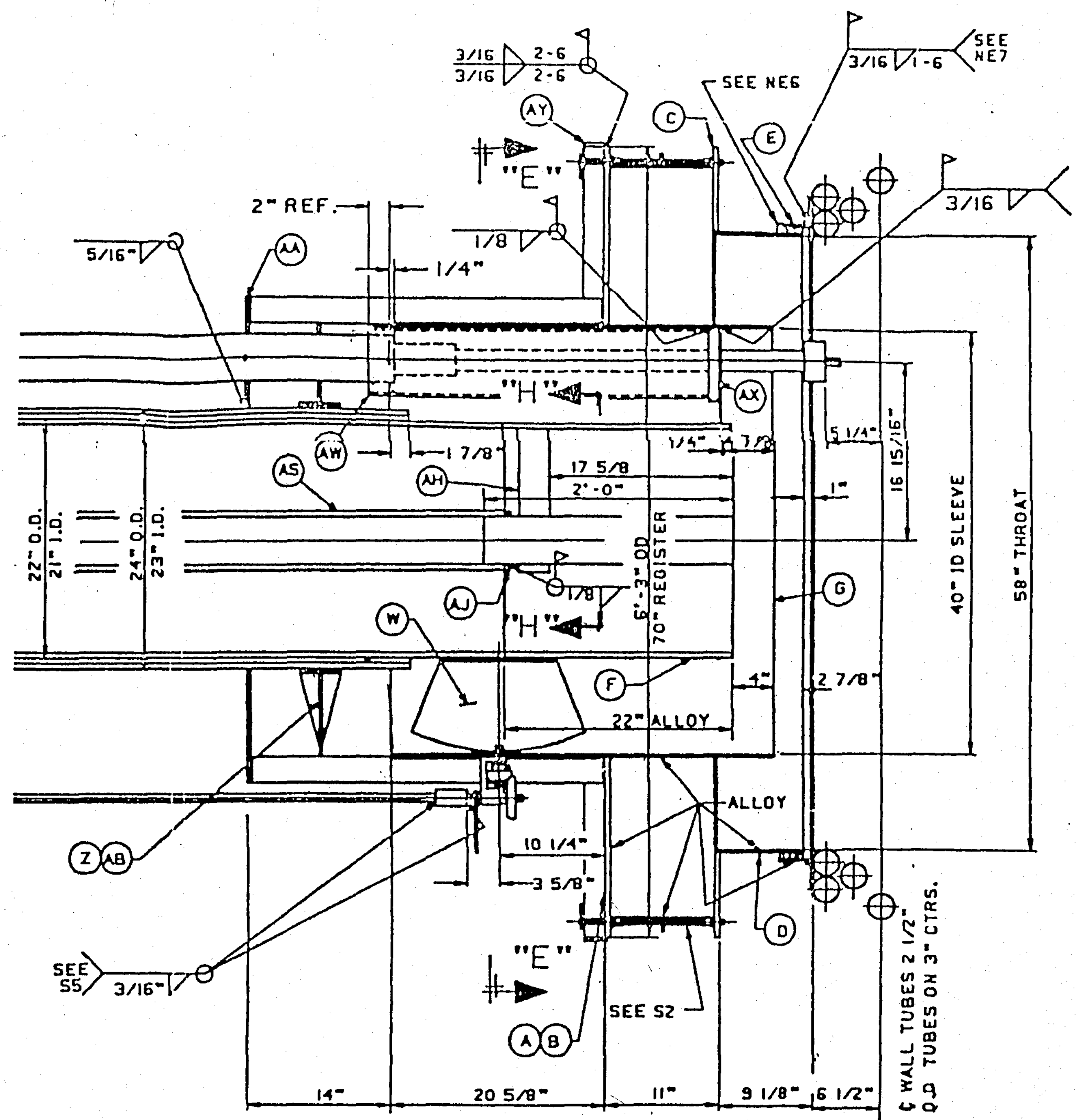

Figure 1 Intermountain Power Project - Existing Design

INTERMOUNTAIN POWER PROJECT
EXISTING DESIGN
(REF. 294361E-12)

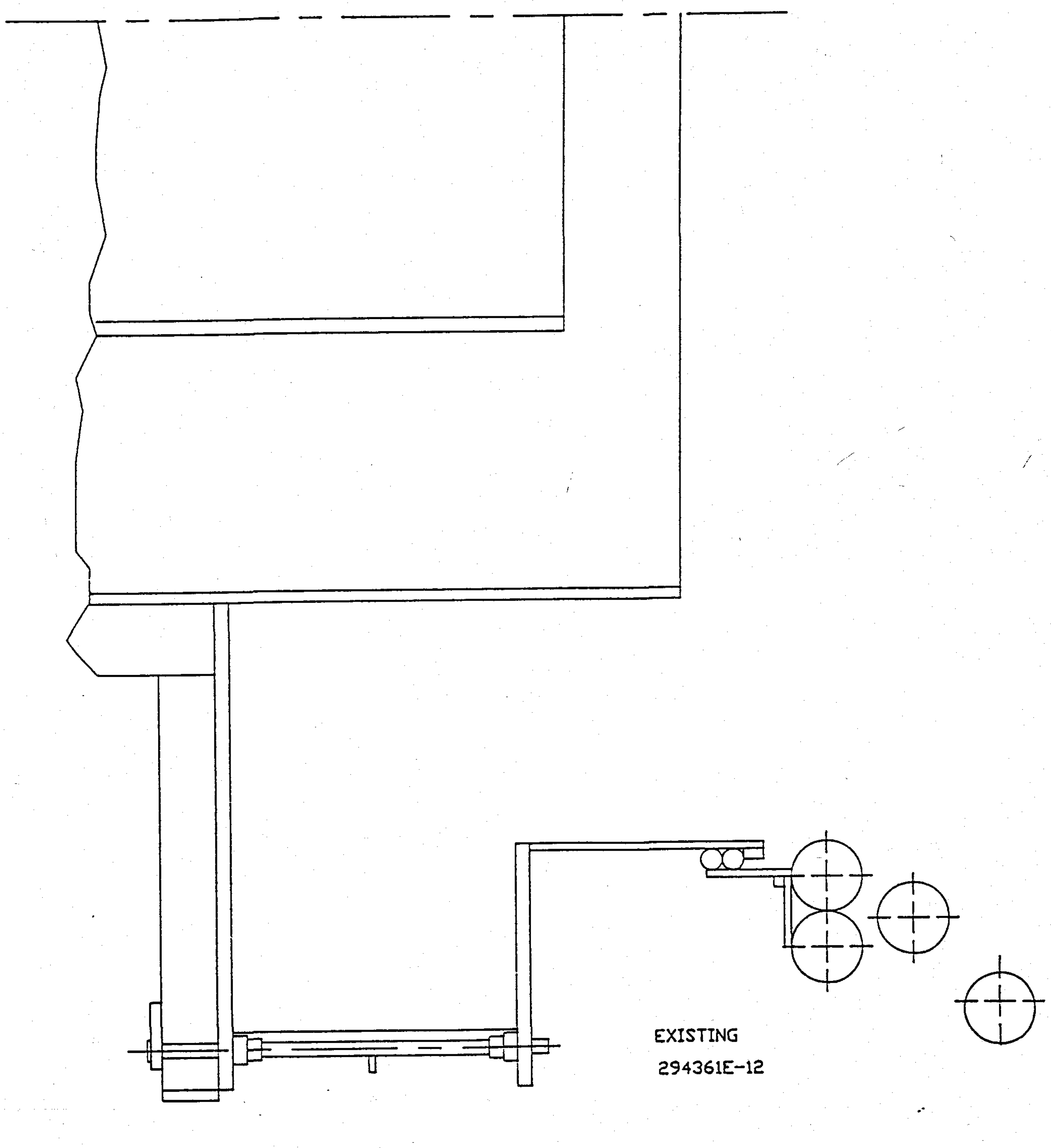

Figure 2 Intermountain Power Project - Existing Design (Expanded view)

INTERMOUNTAIN POWER PROJECT - PROPOSED DESIGN (REF. SK41791E-0)

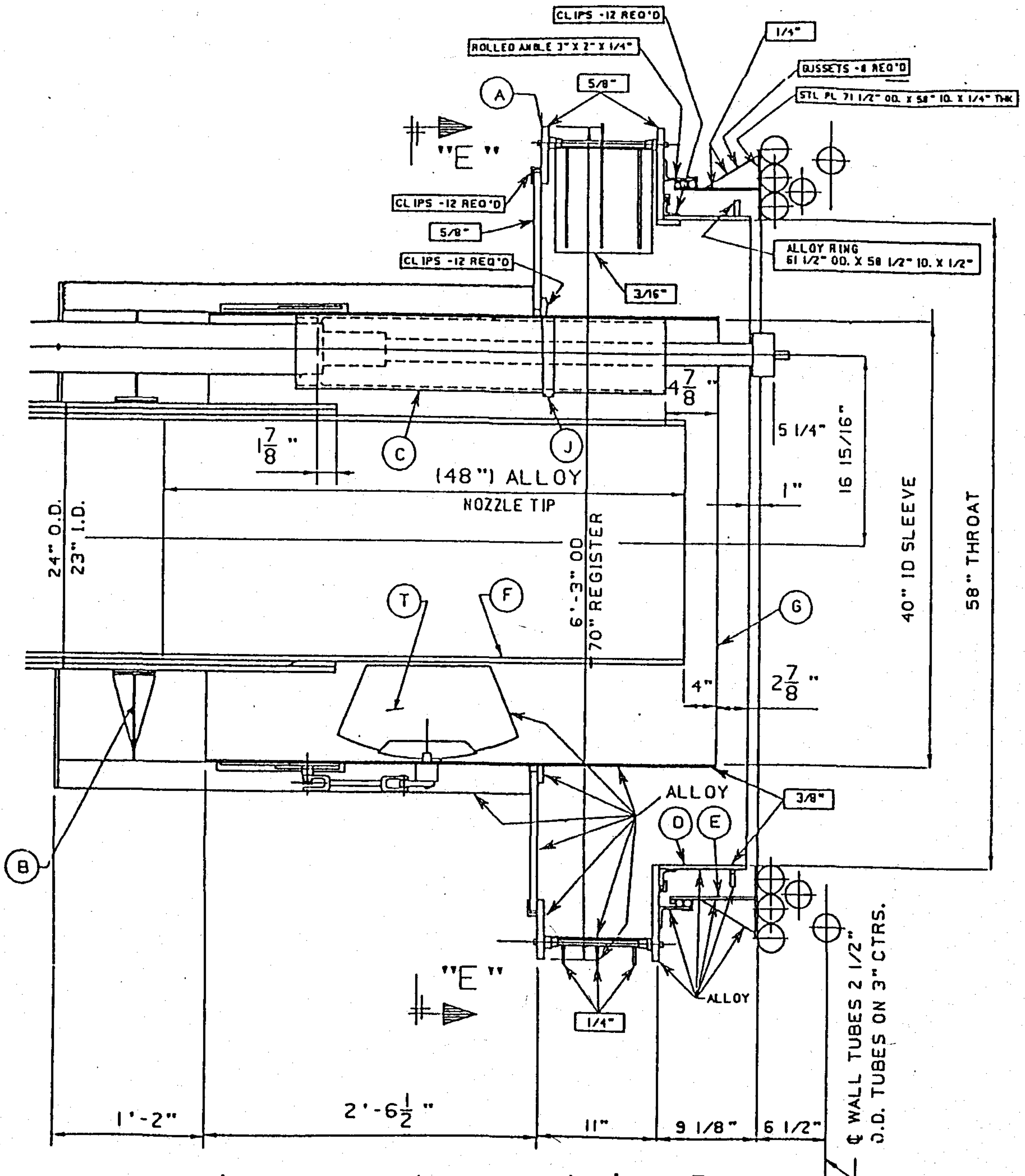

Figure 3 Intermountain Power Project - Proposed Design

INTERMOUNTAIN POWER PROJECT
PROPOSED DESIGN
(REF. SK41791E-0)

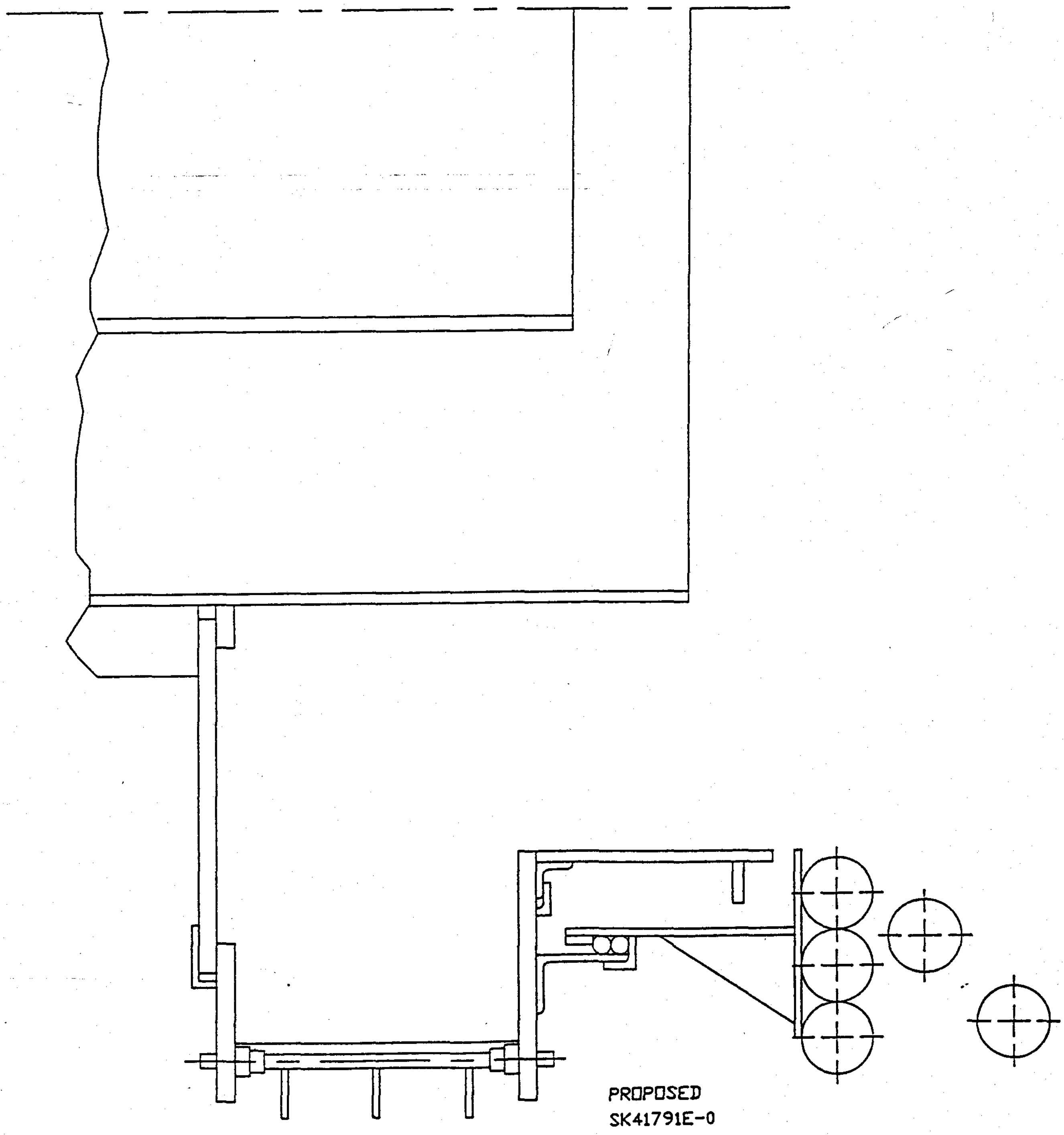

Figure 4 Intermountain Power Project - Proposed Design (Expanded view)

BURNER UPGRADES
PROPOSED MODIFICATIONS SK41791E/0

(REF. RB-614/615 - MAY 1, 1991)

OUTER AIR REGISTER

- o REPLACED WITH MODIFIED HD REGISTER
- o REGISTER FRONT PLATE
  - THICKNESS FROM 1/2" TO 5/8"
  - MATERIAL FROM CARBON STEEL TO 800H
- o REGISTER BACK PLATE
  - THICKNESS FROM 1/2" TO 5/8"
  - MATERIAL FROM TP304 TO 800H
  - SUPPORT LEGS ADDED
  - CENTER SECTION ATTACHED TO FRAME WITH CLIPS (PROVIDES FOR EXPANSION)
- o REGISTER DOOR
  - THICKNESS FROM 10 GA. TO 3/16"
  - ALLOY STIFFENERS ADDED

THROAT SLEEVE

- o THICKNESS FROM 1/4" TO 3/8"
- o MATERIAL FROM TP304 TO 800H
- o EXPANSION RING ADDED TO OD
- o ATTACHED TO FRONT PLATE WITH CLIPS (PROVIDES FOR RADIAL EXPANSION)

Figure 5 Burner Upgrades-
Proposed Modifications

BURNER UPGRADES
PROPOSED MODIFICATIONS SK41791E/0

(REF. RB-614/615 - MAY 1, 1991)

(CONTINUED)

SLIP SEAL

- o MOVED OUTBOARD ON FRONT PLATE TO ELIMINATE INTERFERENCE WITH EXPANSION OF THROAT SLEEVE
- o SEAL REARRANGED TO MINIMIZE RADIANT HEAT ON ROPE PACKING

INNER AIR SLEEVE

- o THICKNESS FROM 1/4" TO 3/8"
- o MATERIAL FROM TP309 TO 800H
- o MATERIAL OF STIFFENERS FROM CARBON STEEL TO 800H
- o SPIN VANE DRIVE OPERATION CHANGED FROM GEARED TO PUSH/PULL
- o INNER SLEEVE LENGTH INCREASED APPROXIMATELY 10"

COAL NOZZLE

- o ALLOY PORTION OF TIP FROM 33" TO 48"

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IP7_004746

Figure 6 Burner Upgrades-Proposed Modifications (continued)

INTERMOUNTAIN POWER PROJECT

RECOMMENDED DESIGN

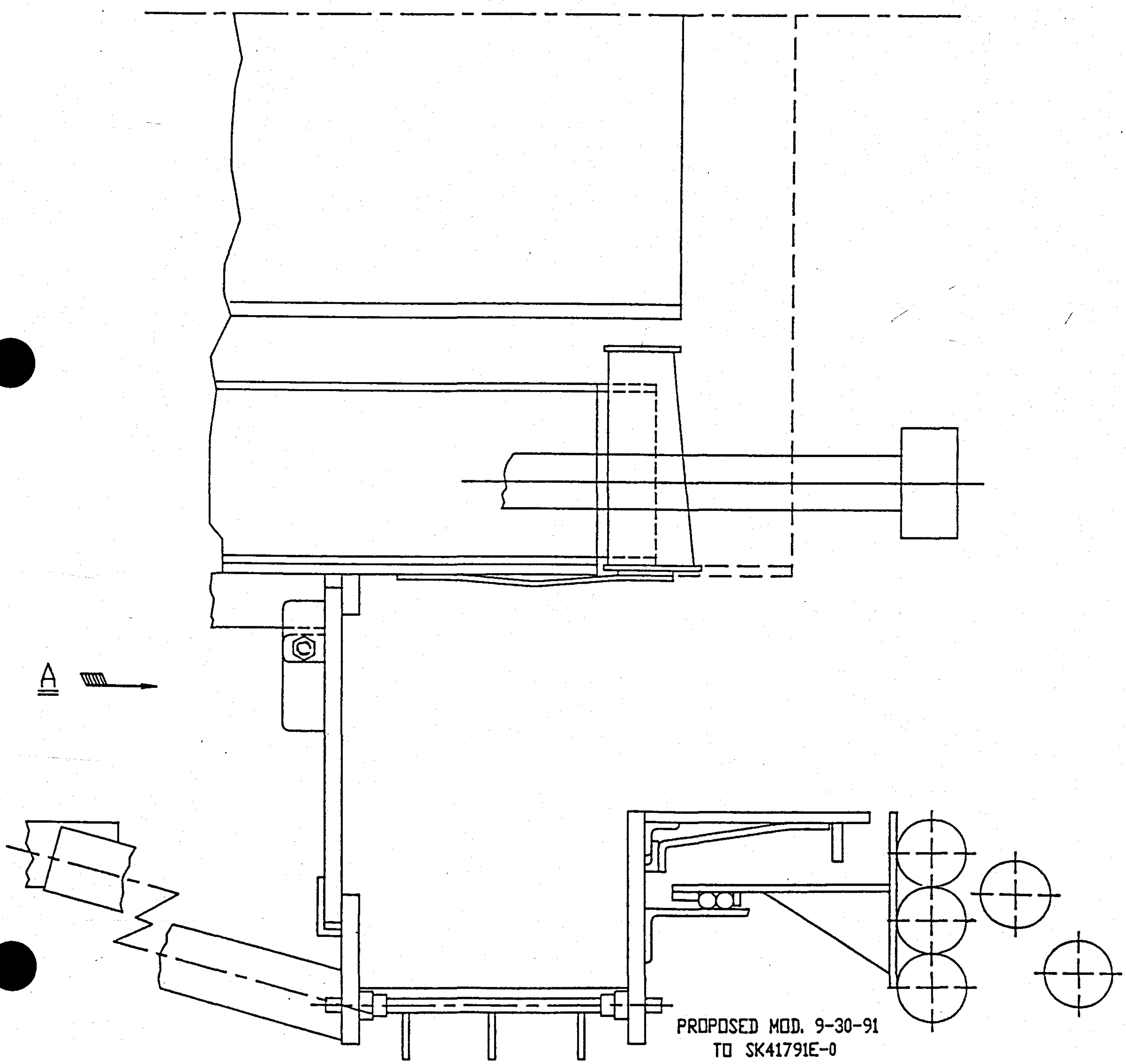

Figure 7 Intermountain Power Project - Recommended Design

EXISTING DESIGN PROBLEMS

COAL PIPE

- o NOZZLE TIP BURNING/WARPING

INNER REGISTER

- o SLEEVE WARPING
- o REGISTER VANE JAMMING

OUTER REGISTER

- o BACK PLATE WARPING
- o THROAT INTERACTION WITH AIR SEAL
- o REGISTER VANE JAMMING

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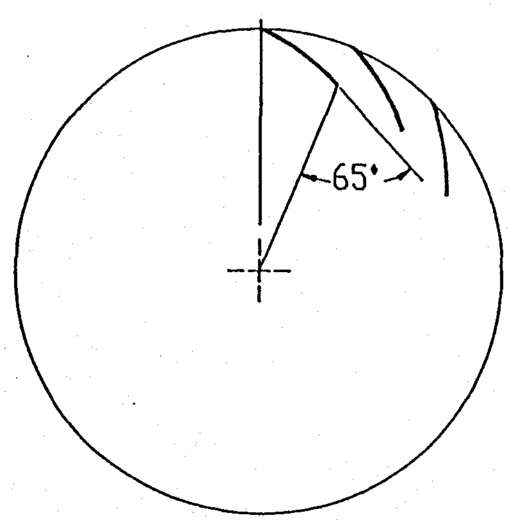
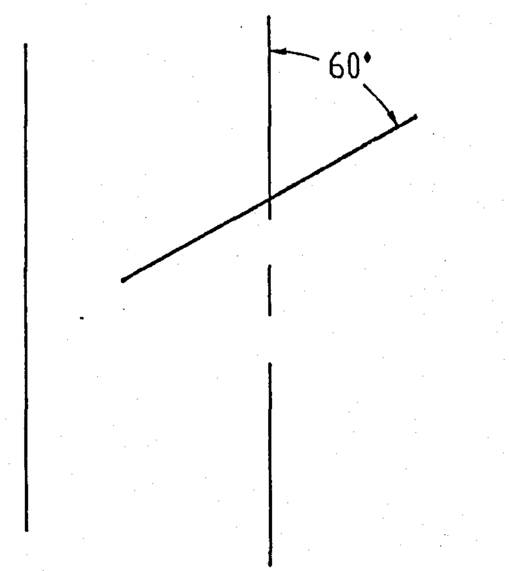
Figure 8 Existing Design Problems

AIR REGISTERS

- o OPERATING CONDITIONS
  - 100 PERCENT LOAD WITH 42 BURNERS
  - AIR TEMPERATURE = 650°F
  - WINDBOX TO FURNACE DIFFERENTIAL PRESSURE = 2.0 INCHES WATER
- o SETTINGS (REF. NOVEMBER 24, 1988)



OUTER REGISTER VANE
EXIT FLOW ANGLE (OFF RADIAL)



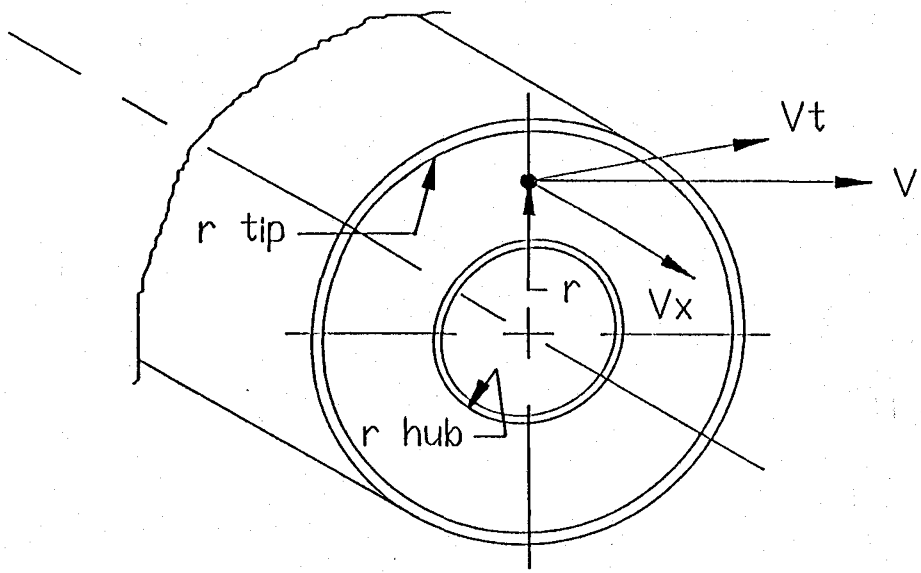
INNER SPIN VANE
EXIT FLOW ANGLE (OFF AXIAL)

IPP.AR

Figure 9 Air Registers

SWIRL NUMBER

- o MEASURE OF JET TANGENTIAL TO AXIAL MOMENTUM
- o DETERMINES SIZE OF COMBUSTION INTERNAL RECIRCULATION ZONE



$$\text{Local Swirl No.} = \frac{rV_t}{V_x\, r_{tip}}$$

$$\text{Integrated Swirl No.} = \frac{1}{r_{tip}} \frac{\int_{r_{hub}}^{r_{tip}} rV_t \; (\rho V_x \; 2\pi r)\,dr}{\int_{r_{hub}}^{r_{tip}} V_x \; (\rho V_x \; 2\pi r)\,dr}$$

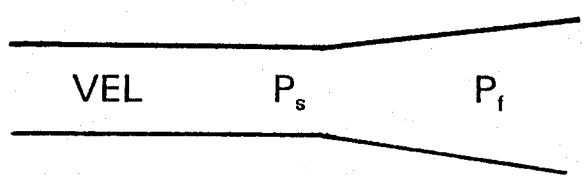
IPP.SN(COR)

Figure 10 Swirl Number

IP7_004750

RECIRCULATION PARAMETER

- MEASURE OF AXIAL MOMENTUM TO OVERCOME LOCAL STATIC PRESSURE TO FURNACE PRESSURE RISE
- POTENTIAL FOR RECIRCULATION EXISTS WHEN THE PARAMETER IS A NEGATIVE VALUE



$(P_s < P_f)$

RECIRCULATION PARAMETER = [(AXIAL MOMENTUM / UNIT AREA) - PRESSURE RISE]

$$\text{RECIRCULATION PARAMETER} = [(\rho V^2 / g_c) - (P_f - P_s)]$$

IPP.RP

Figure 11 Recirculation Parameter

INTER MOUNTAIN POWER PROJECT - UNITS 1 & 2

OUTER AIR REGISTER - EXISTING DESIGN

AIR FLOW

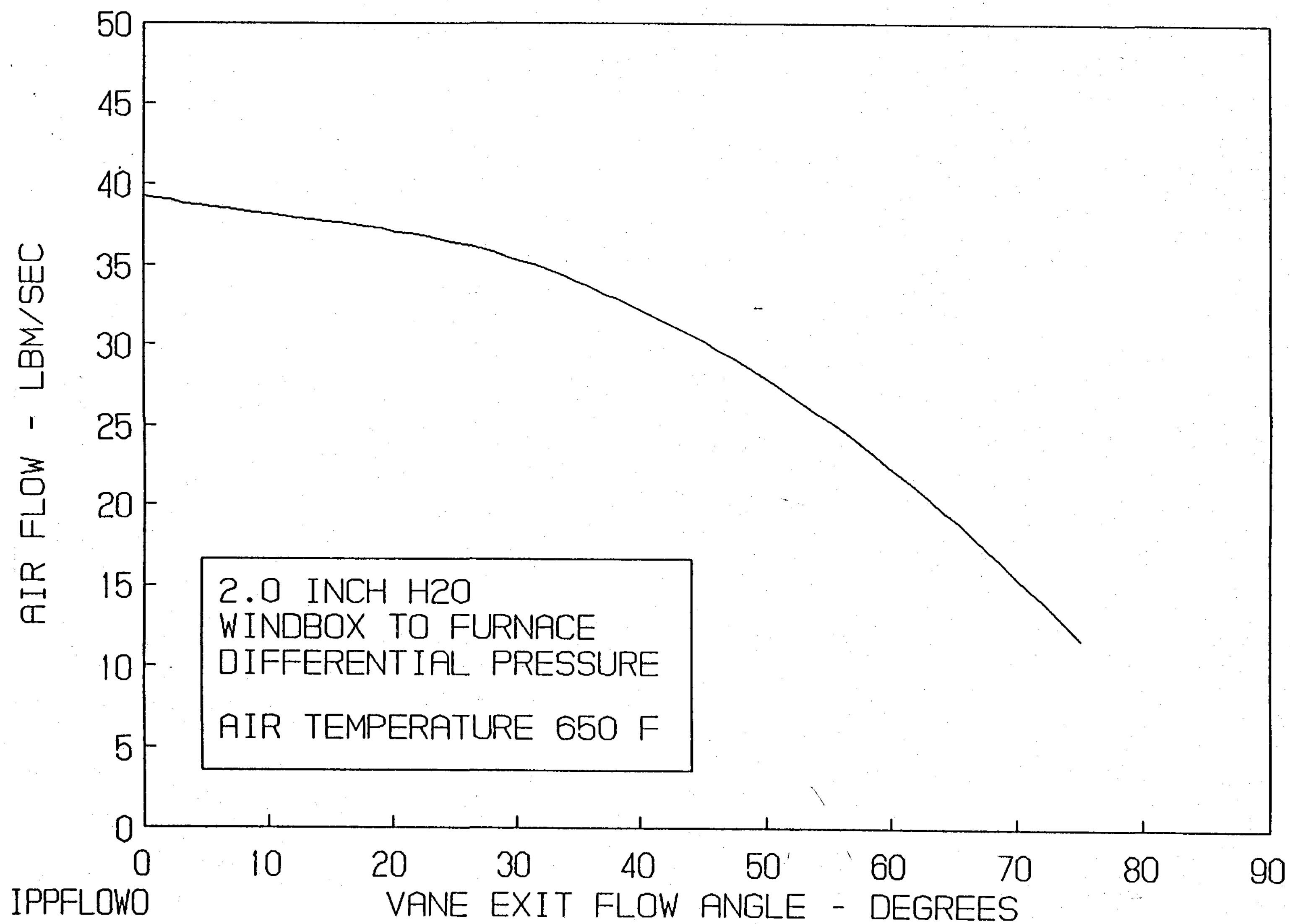


**Figure 12 Outer Air Register-
Existing Design Air Flow**

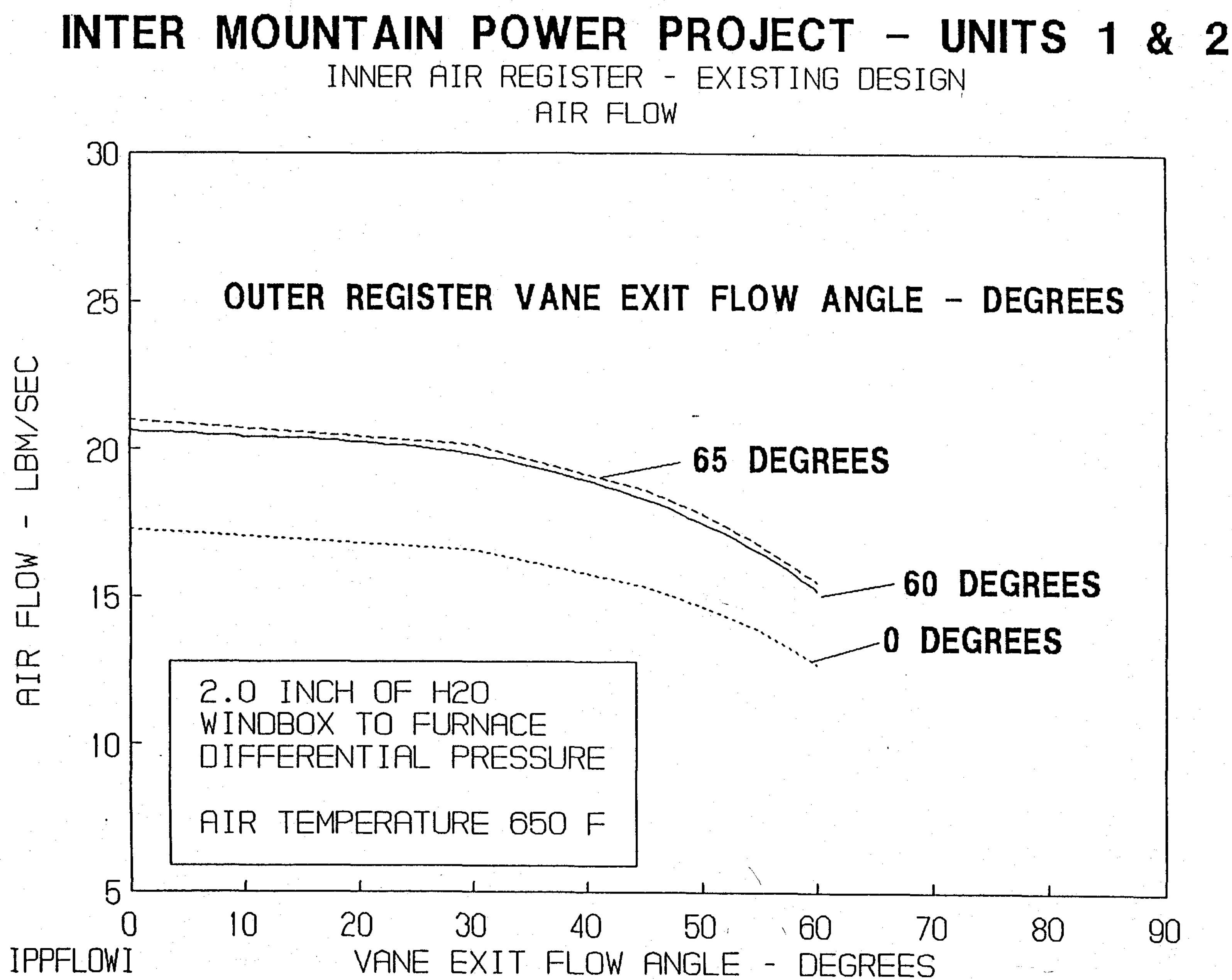


Figure 13 Inner Air Register-
Existing Design Air Flow

IP7_0004753

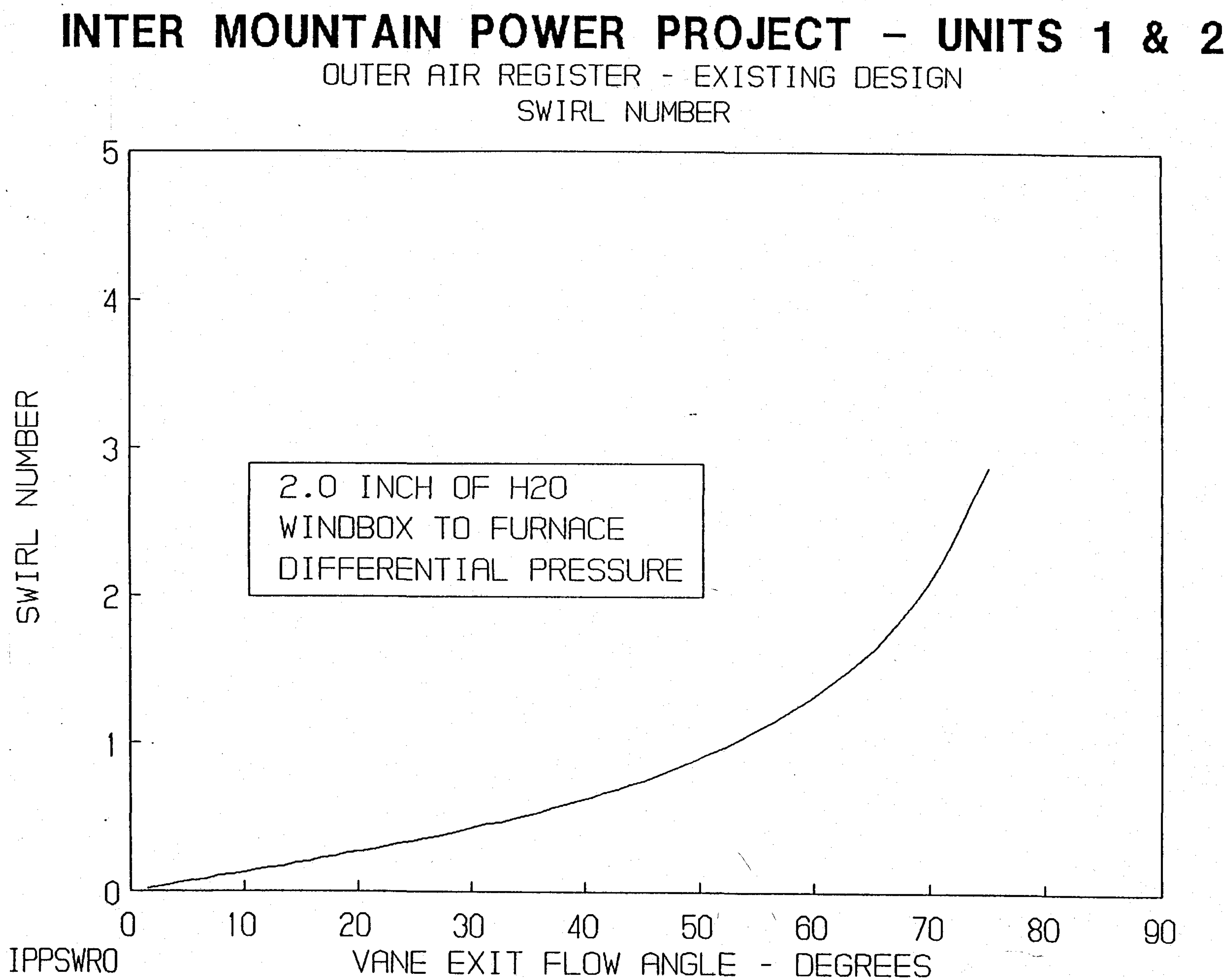


Figure 14 Outer Air Register- Existing Design Swirl Number

INTER MOUNTAIN POWER PROJECT - UNITS 1 & 2

INNER AIR REGISTER - EXISTING DESIGN
SWIRL NUMBER

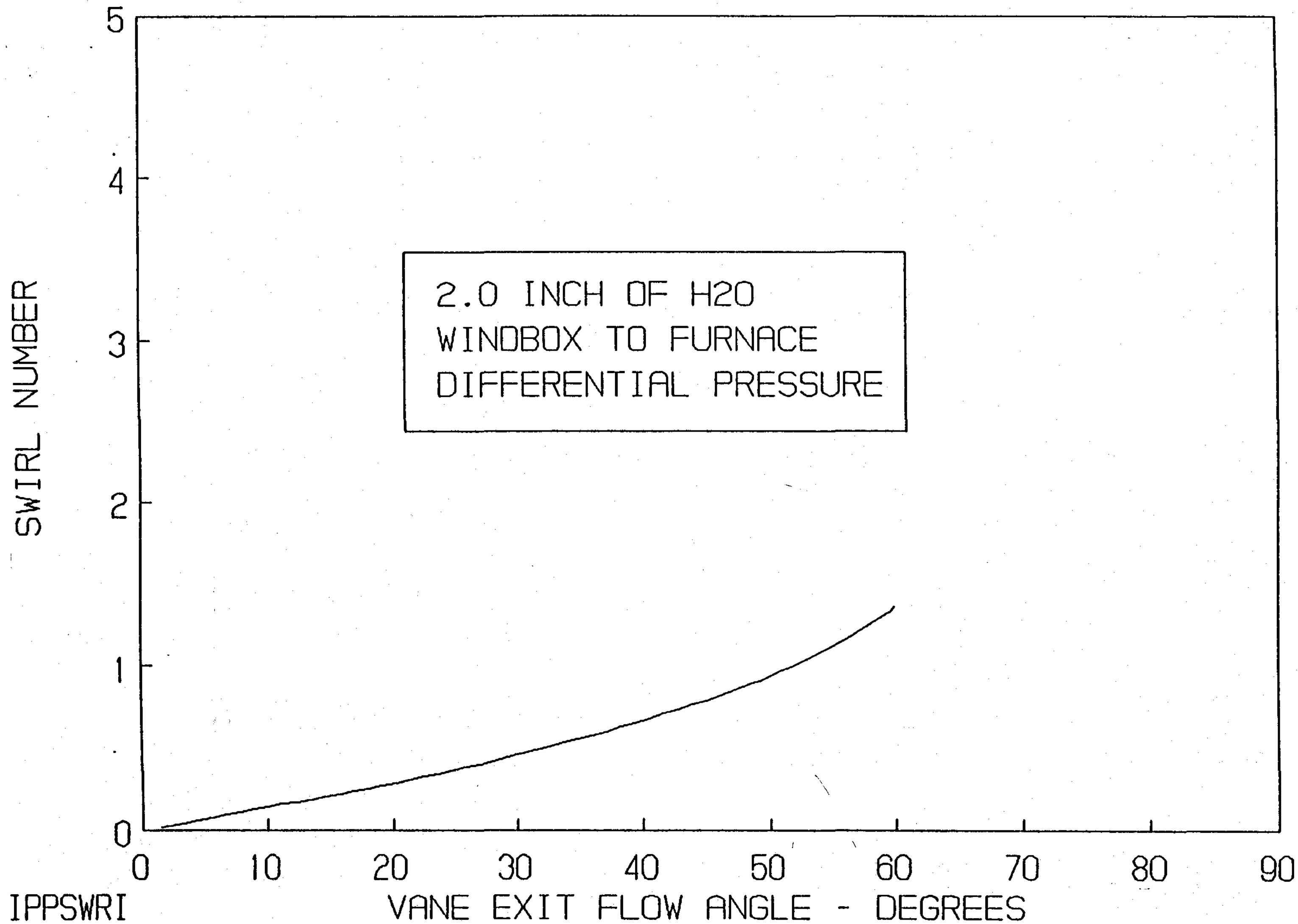


Figure 15 Inner Air Register- Existing Design Swirl Number

IP7_004755

INTER MOUNTAIN POWER PROJECT - UNITS 1 & 2

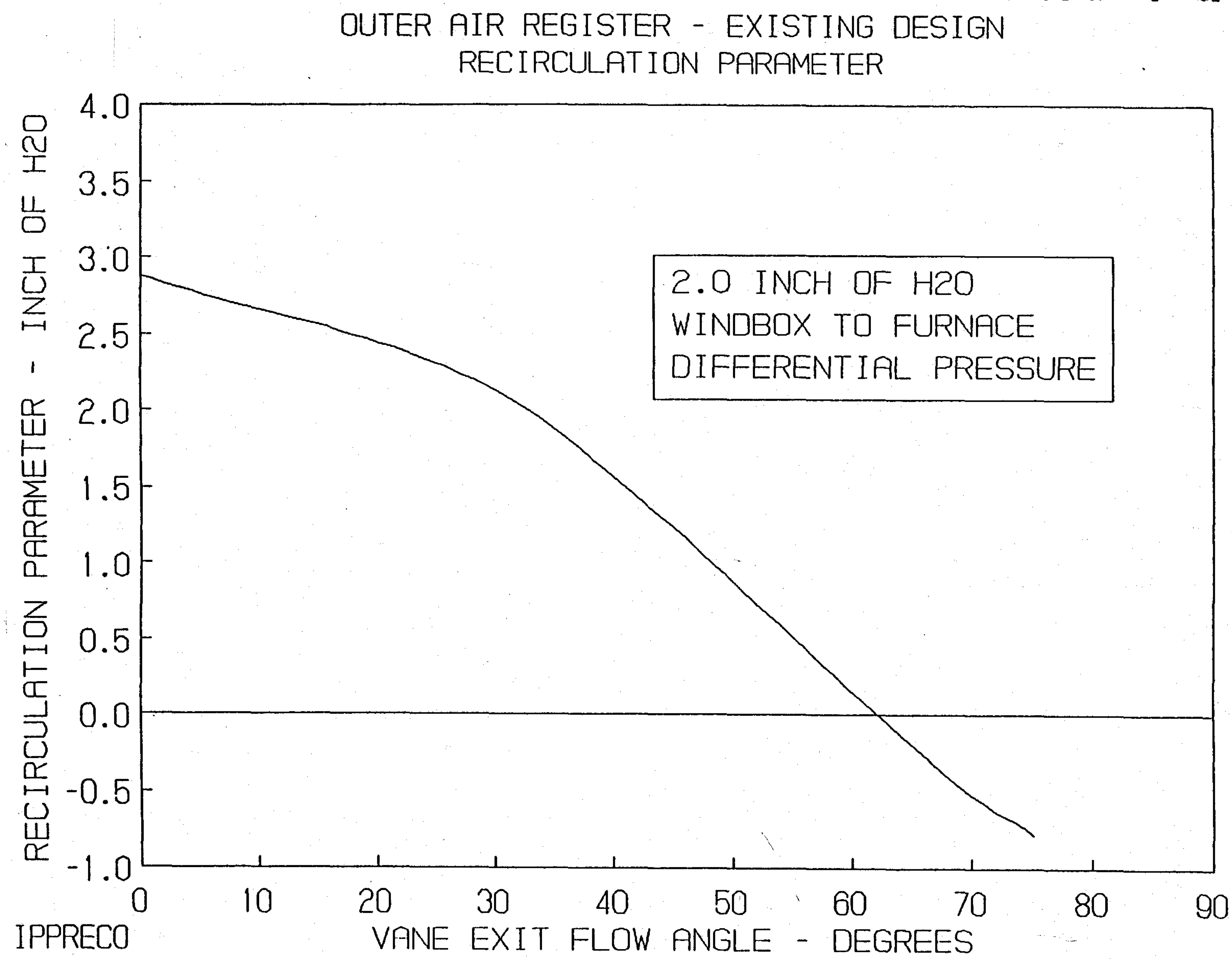


Figure 16 Outer Air Register-
Existing Design
Recirculation Parameter

IP7_004756

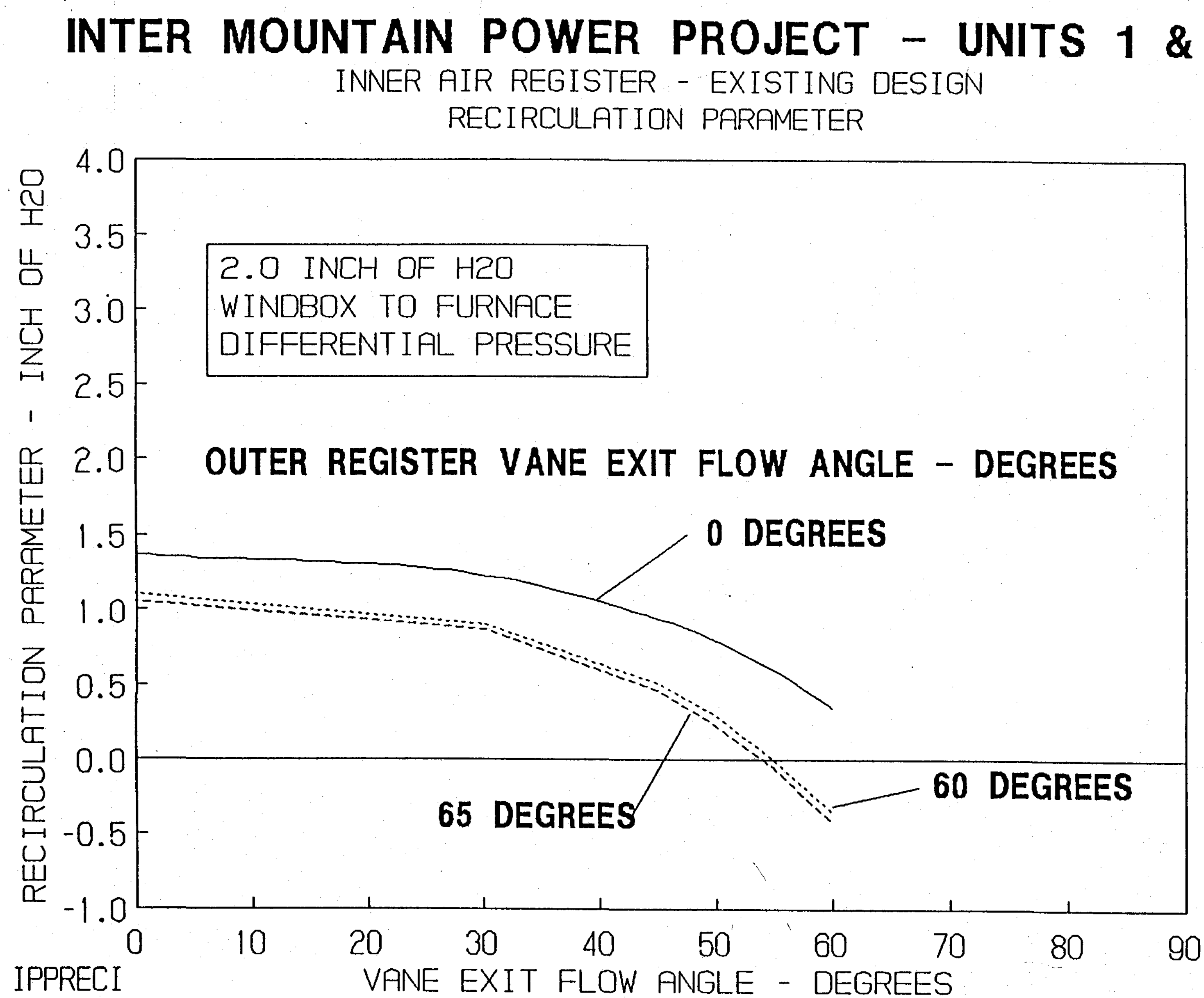


Figure 17 Inner Air Register-
Existing Design
Recirculation Parameter

BURNER - UNIT 2

(REF. RB-0615 - NOVEMBER 24, 1988)

BURNER MODIFICATIONS

- o EXPANSION JOINTS WERE INSTALLED ON THE OUTER REGISTER DRIVE HANDLES
- o BACKPLATES AMD FRONT PLATES OF REGISTERS WERE CUT FREE AND EXPANSION CLIPS WERE INSTALLED

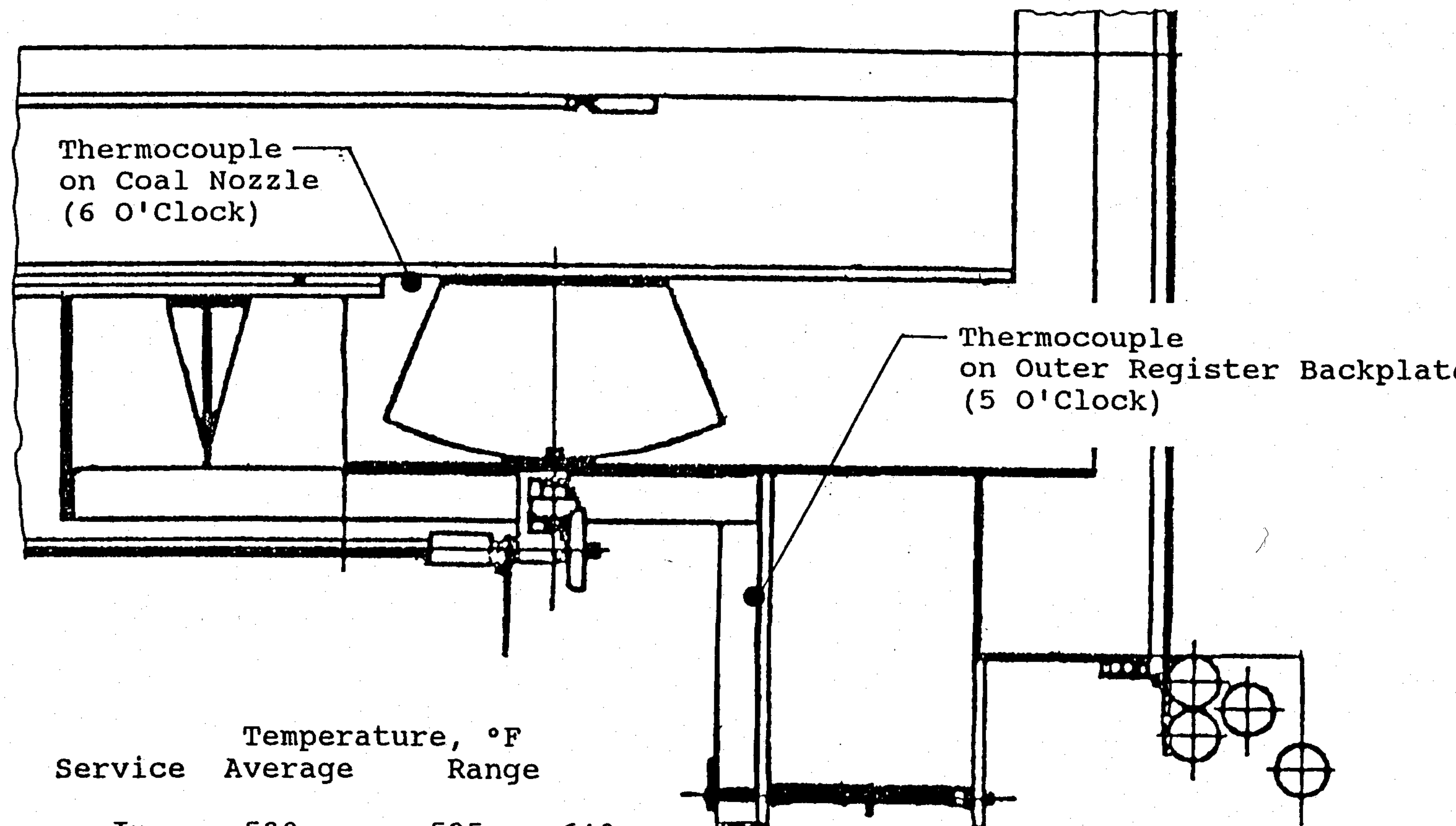
BURNER SETTINGS

- o OUTER REGISTERS - 6" (DOOR STIFFENER TO DOOR - ON A PERPENDICULAR)
- o SPIN VANES - 30° (WHERE 90° IS STRAIGHT THROUGH, 0° IS CLOSED)
- o BACK PLATES - 5", 4", 3", 3", 4", 5" OPEN
- o (ALL BURNER SETTINGS HAVE BEEN LOCKED IN PLACE)

IPP.BS

Figure 18 Burner - Unit 2
(November 24, 1988)

MEASURED TEMPERATURES AT FULL LOAD
(AUGUST 30, 1991)



	Service	Temperature, °F Average	Range	
Coal Nozzle	In	580	525 - 640	
	Out	980	830 - 1200	
Outer Register (Back Side)	In	1000	850 - 1220	
	Out	1175	980 - 1285	(1450°F MAX)

Figure 19 Measured Temperatures at Full Load (August 30, 1991)

INTERMOUNTAIN POWER PROJECT
HEAT TRANSFER ANALYSIS

IN SERVICE

- o CONVECTIVE COOLING FROM 100 PERCENT LOAD REGISTER AIRFLOW
- o FLAME RADIATION PRIMARILY ON BACK WALL
- o BACK PLATE TEMPERATURE: 1,050°F (ANALYSIS) VS. 1,000°F AVG (THERMOCOUPLE)

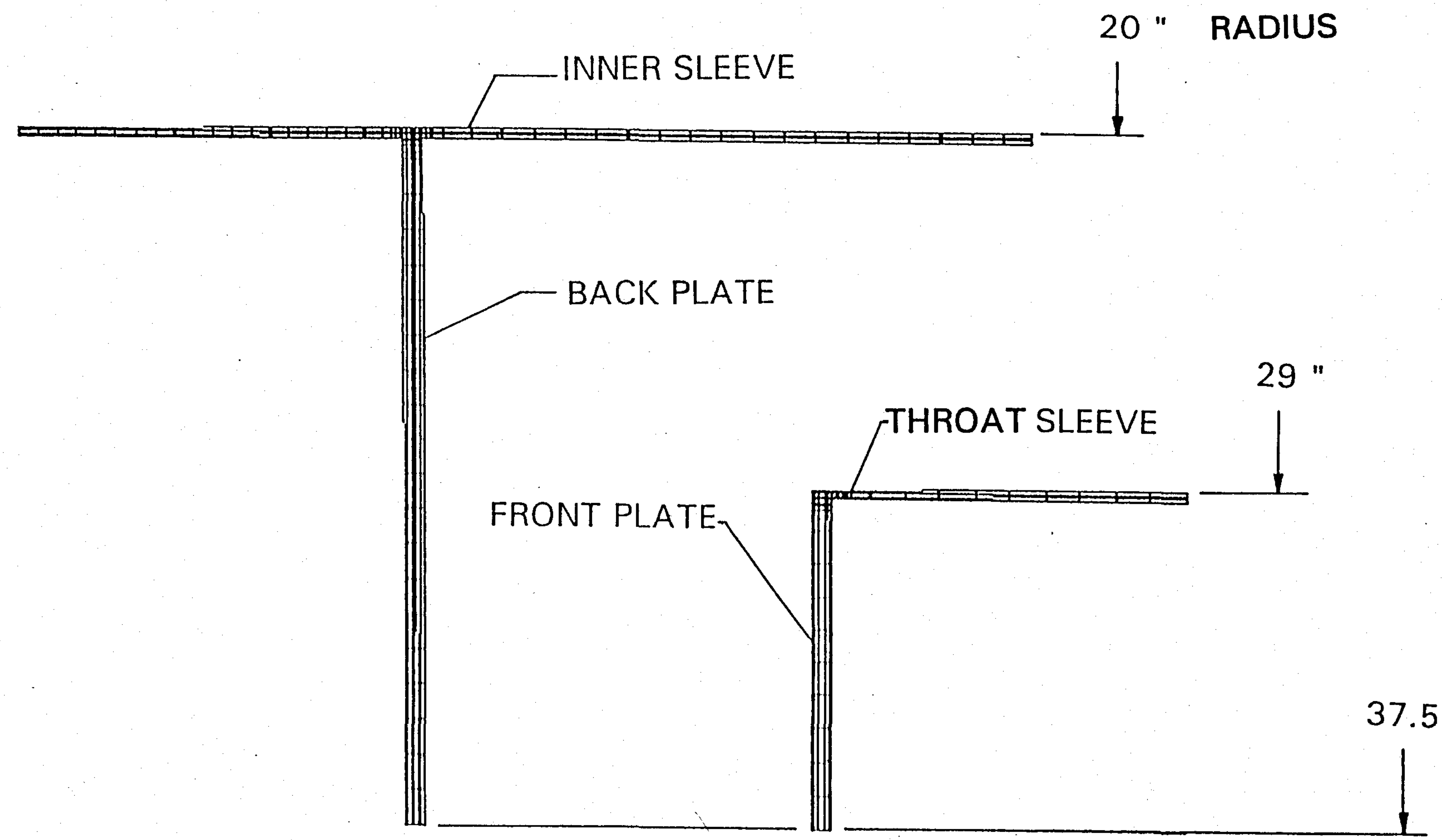
OUT OF SERVICE

- o CONVECTIVE COOLING TO 20 PERCENT AIRFLOW
- o RADIATION LOAD SAME AS IN SERVICE FOR WORST CASE ANALYSIS
- o ACTUAL RADIATION IS LESS WITH BURNER FLAME OUT
- o PREDICTED BACK PLATE TEMPERATURE HIGHER THAN MEASURED

IPPHTA

Figure 20 Heat Transfer Analysis
Design Conditions

FINITE ELEMENT MODEL: EXISTING DESIGN



IP7_004761

Figure 21 Finite Element Model:
Existing Design

FINITE ELEMENT MODEL: MODIFIED BACKPLATE

FOUR SEGMENT PANEL

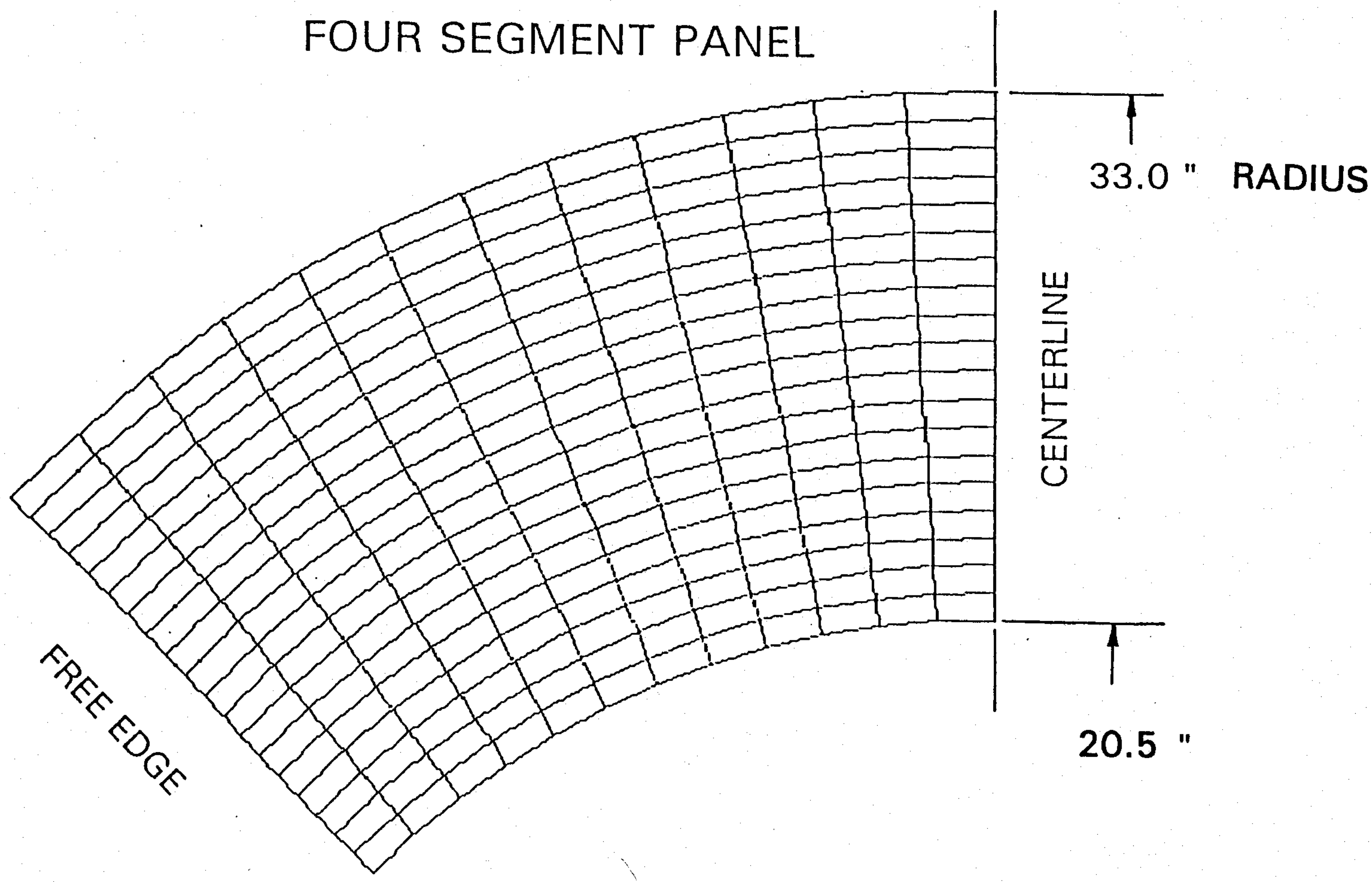


Figure 22 Finite Element Model:
Modified Backplate

ALLOWABLE STRESS VS. TEMPERATURE

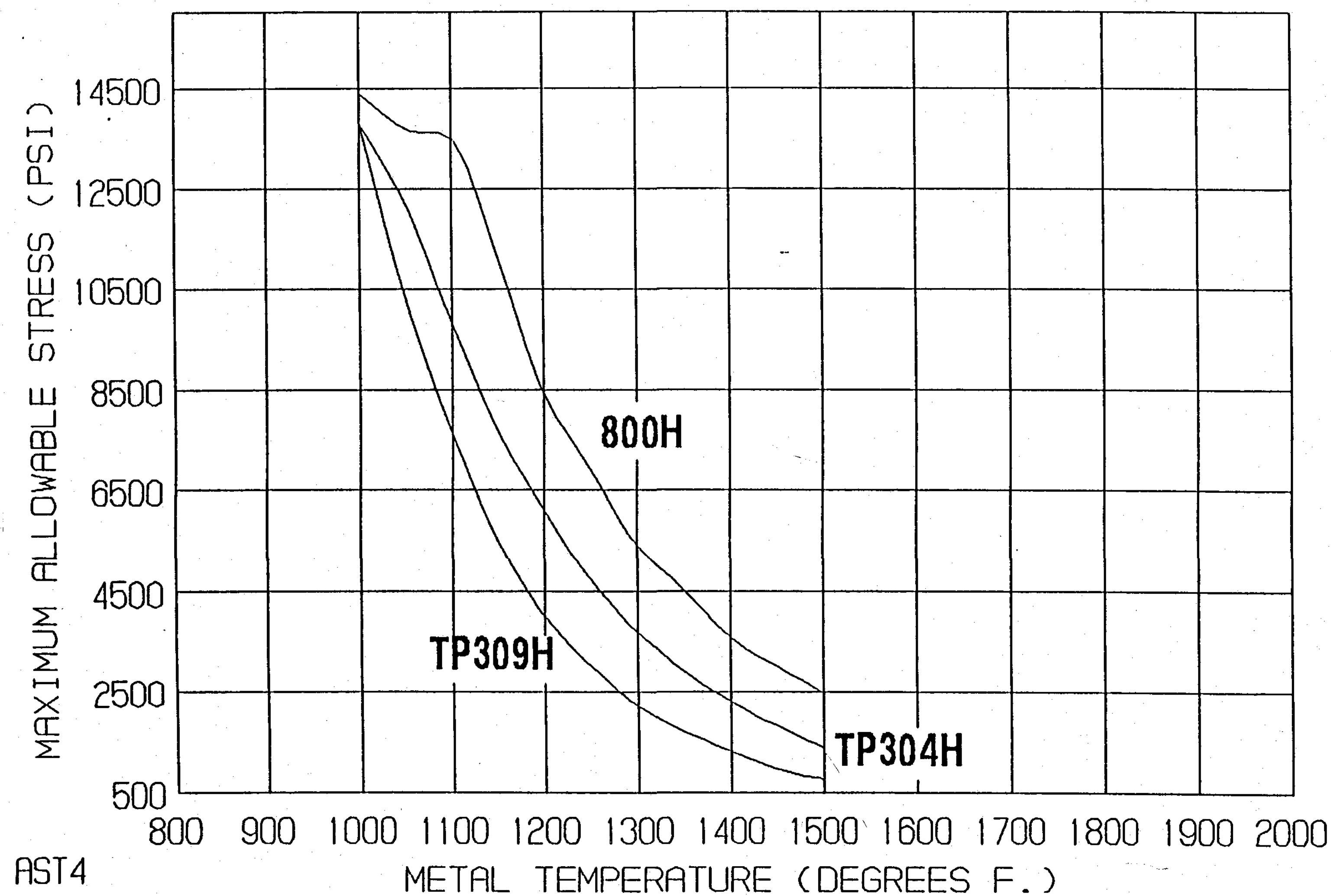


Figure 23 Allowable Stress vs. Temperature

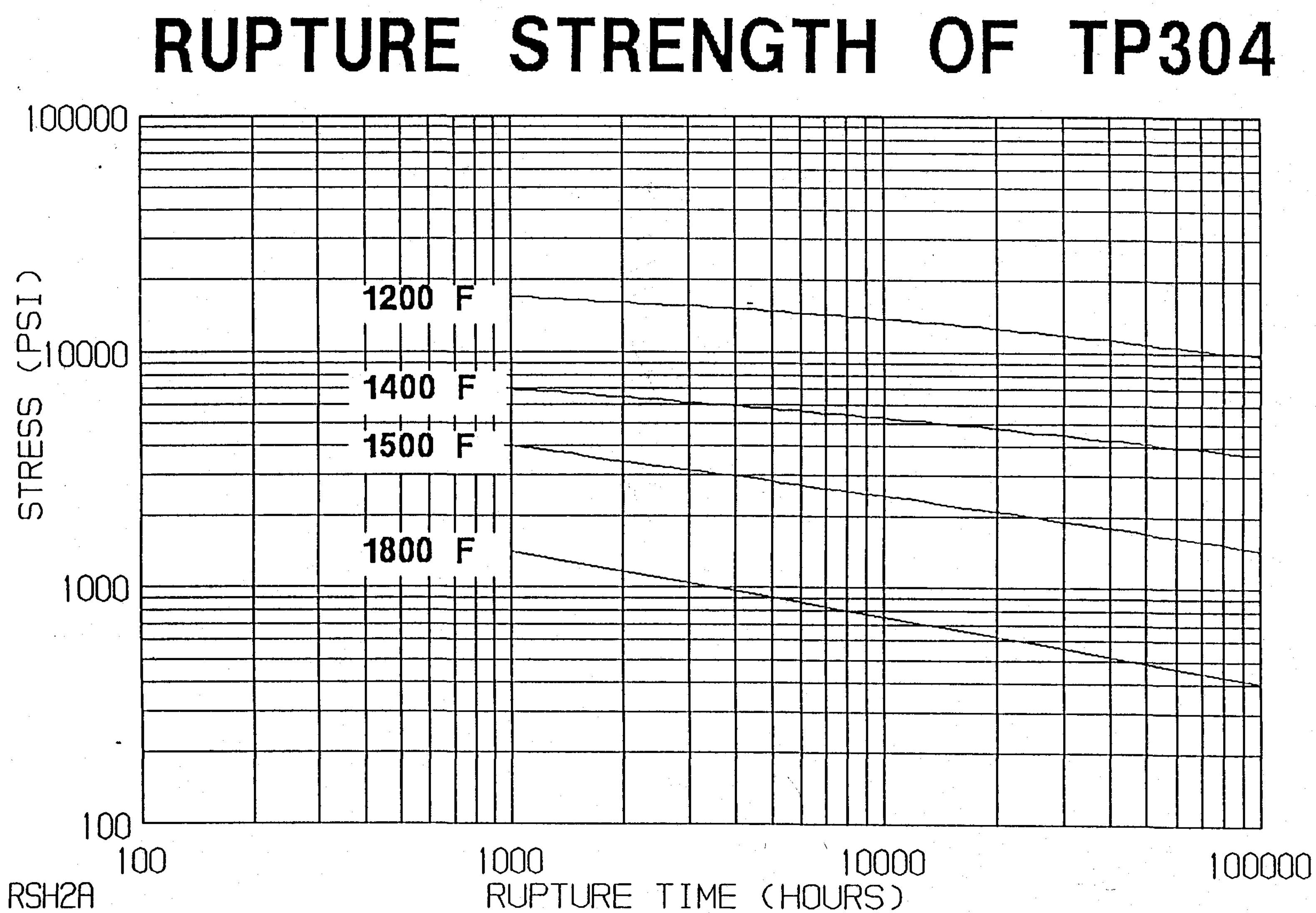


Figure 24 Rupture Strength of TP304

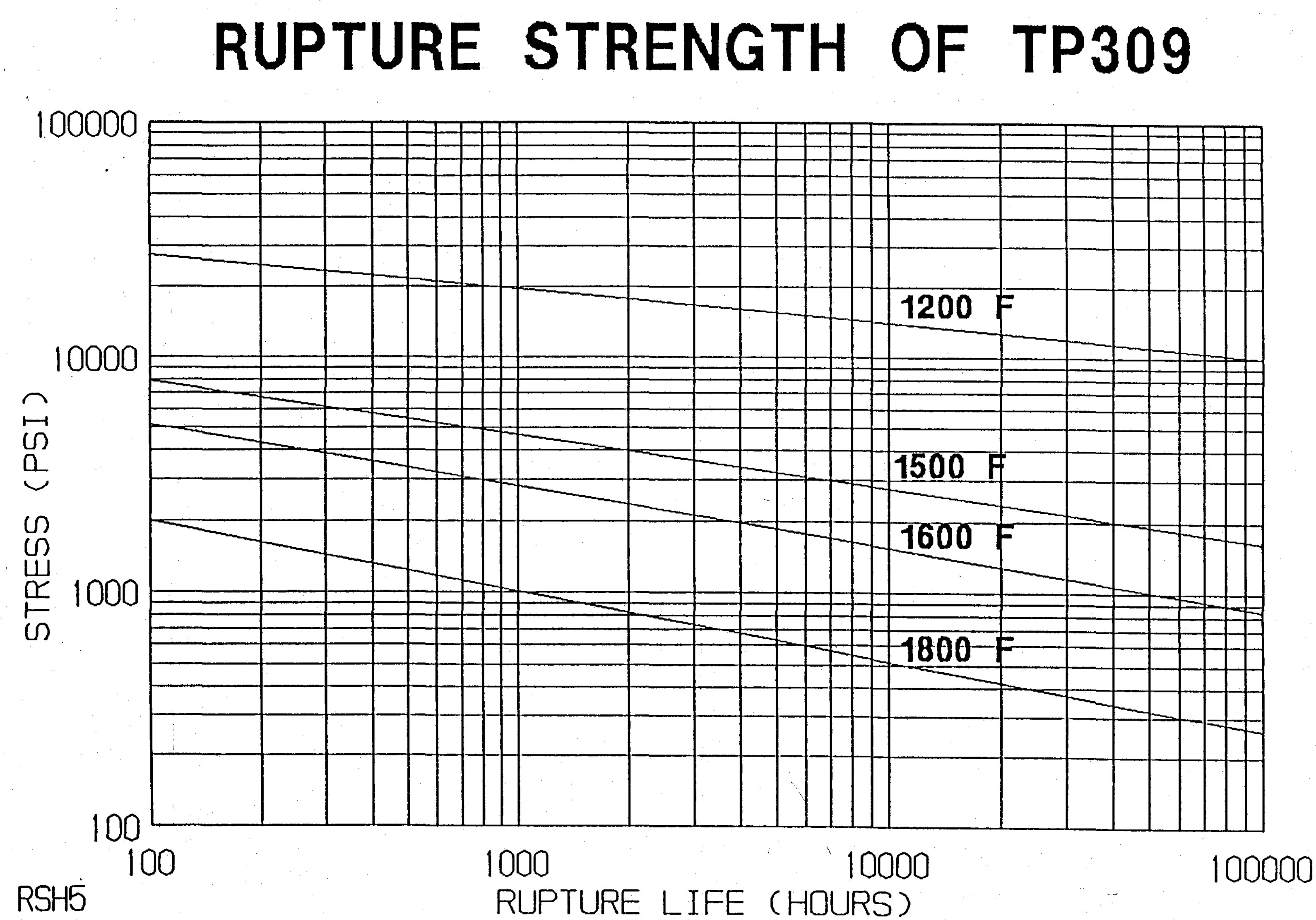


Figure 25 Rupture Strength of TP309

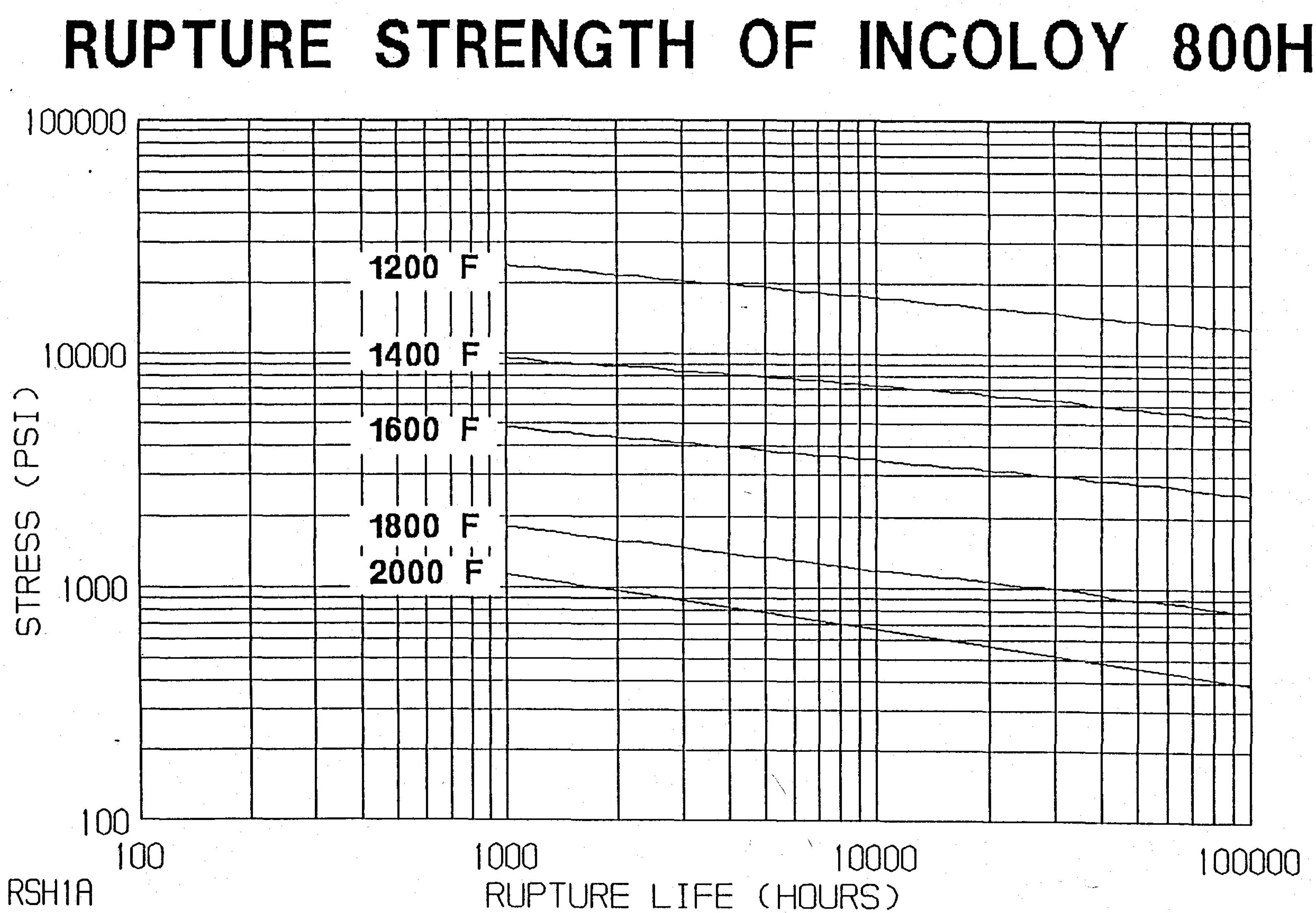


Figure 26 Rupture Strength of Incoloy 800 HT

INTERMOUNTAIN POWER PROJECT
EXISTING DESIGN : IN SERVICE
HEAT TRANSFER ANALYSIS

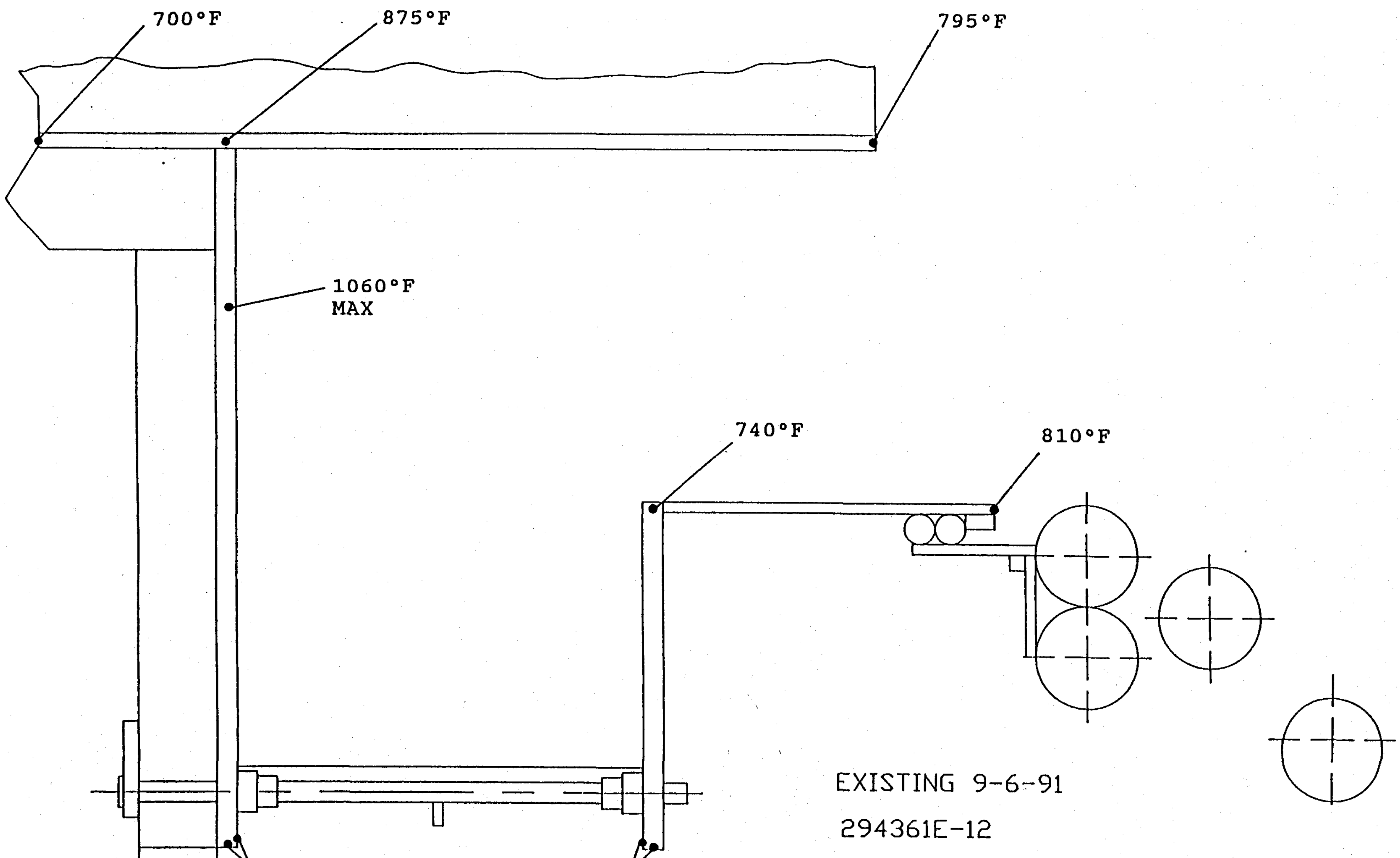


Figure 27 Existing Design: In Service Heat Transfer Analysis

INTERMOUNTAIN POWER PROJECT
EXISTING DESIGN: IN SERVICE
FINITE ELEMENT MODEL:
DEFORMATION ANALYSIS

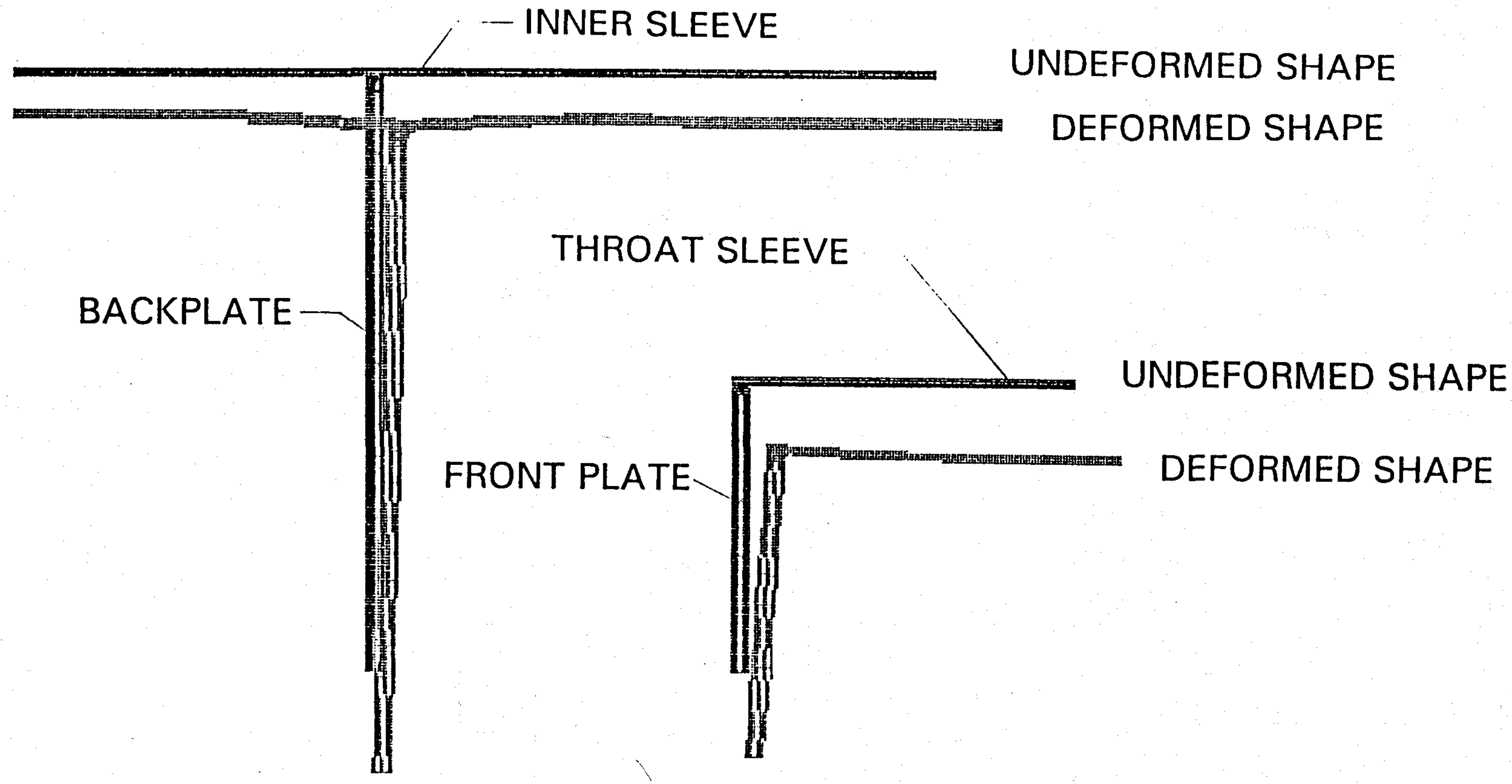
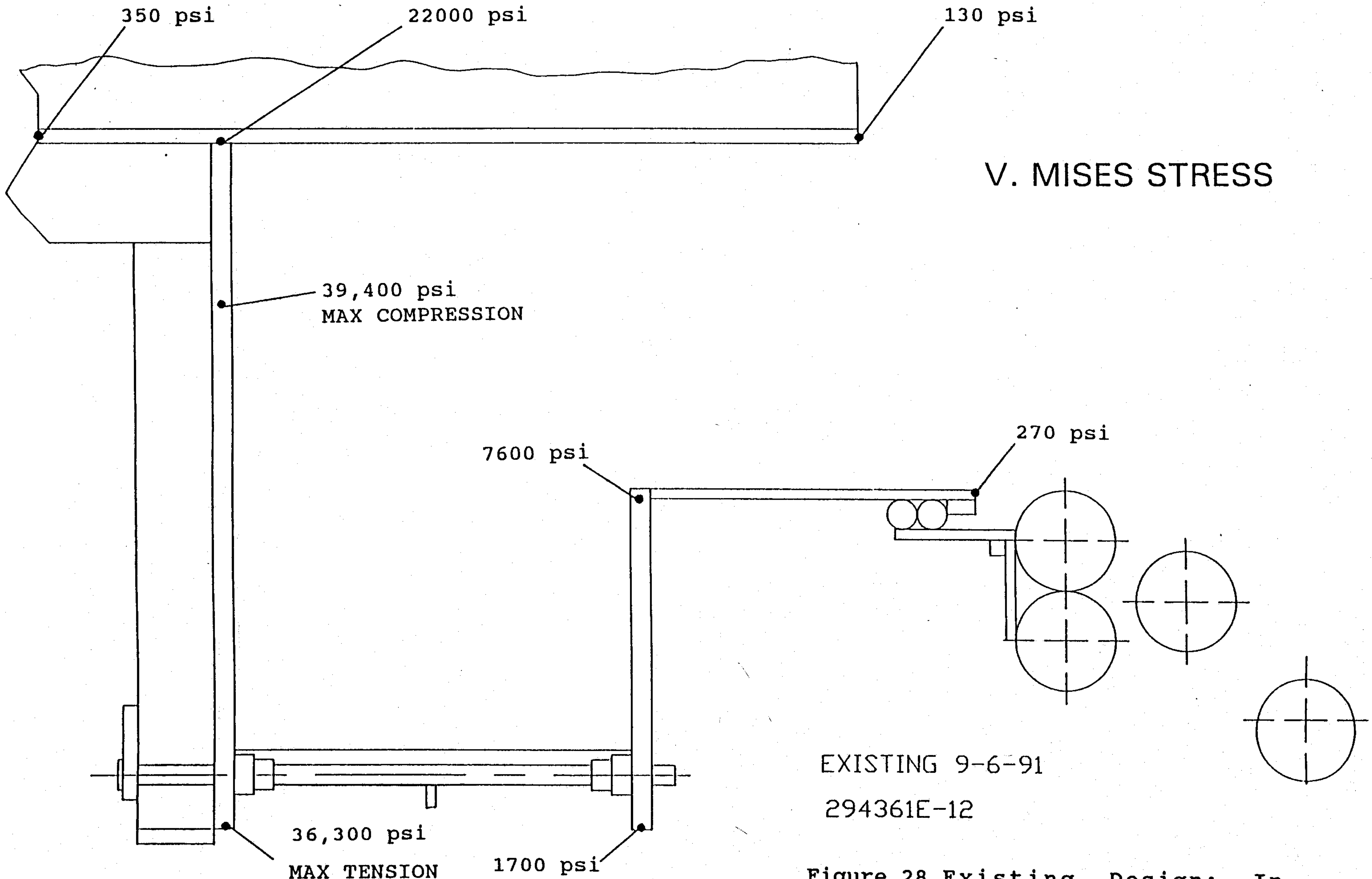


Figure 27A Existing Design: In Service
Deformation Analysis

INTERMOUNTAIN POWER PROJECT
EXISTING DESIGN : IN SERVICE
STRESS ANALYSIS



IP7_004769

Figure 28 Existing Design: In

SUMMARY
EXISTING DESIGN (REF. 294361-12)
IN SERVICE

BACK PLATE

- o HOT SPOT ON BACK PLATE OUTBOARD OF INNER SLEEVE WITHIN RANGE OF MEASURED TEMPERATURES
- o HIGH RADIAL TEMPERATURE GRADIENT CAUSES HIGH TANGENTIAL STRESS GRADIENT
- o SEPARATION FROM SLEEVE PREDICTED WITH SUBSEQUENT CONING/BUCKLING

INNER SLEEVE

- o LOW TEMPERATURE AND STRESSES EXCEPT FOR LOCAL STRESS CONCENTRATION AT BACK PLATE ATTACHMENT
- o TEMPERATURE/STRESS RELIEF WILL OCCUR FOLLOWING SEPARATION OF BACK PLATE

FRONT PLATE & THROAT SLEEVE

- o LOW TEMPERATURE AND STRESSES EXCEPT FOR LOCAL STRESS CONCENTRATION AT JOINT
- o PREDICTED PEAK STRESS AT JOINT WITHIN ALLOWABLE LIMITS

IPPSUM1

Figure 29 Summary Existing Design: In Service

INTERMOUNTAIN POWER PROJECT
EXISTING DESIGN : OUT OF SERVICE
HEAT TRANSFER ANALYSIS

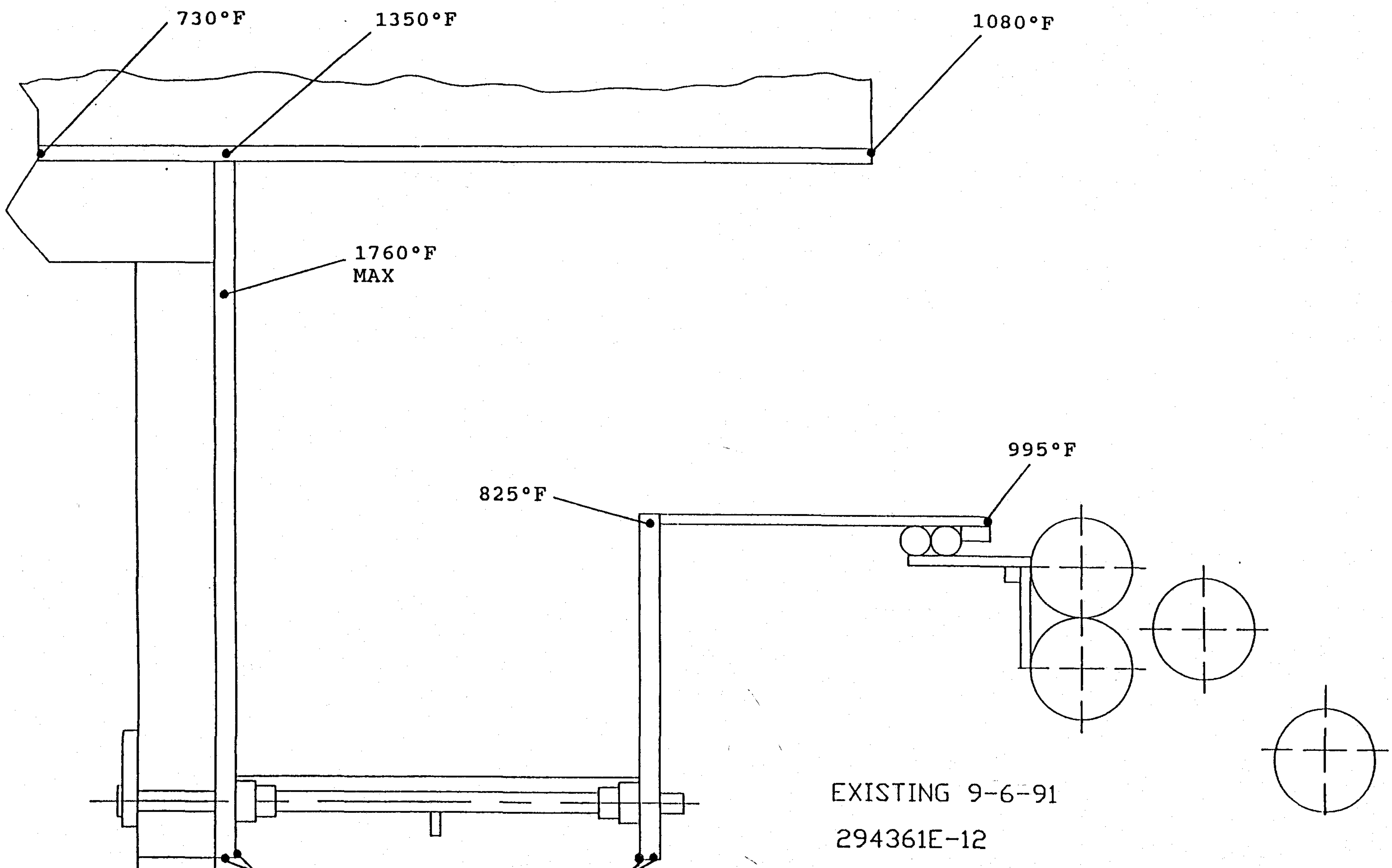


Figure 30 Existing Design: Out of Service Heat Transfer

INTERMOUNTAIN POWER PROJECT
EXISTING DESIGN : OUT OF SERVICE
STRESS ANALYSIS

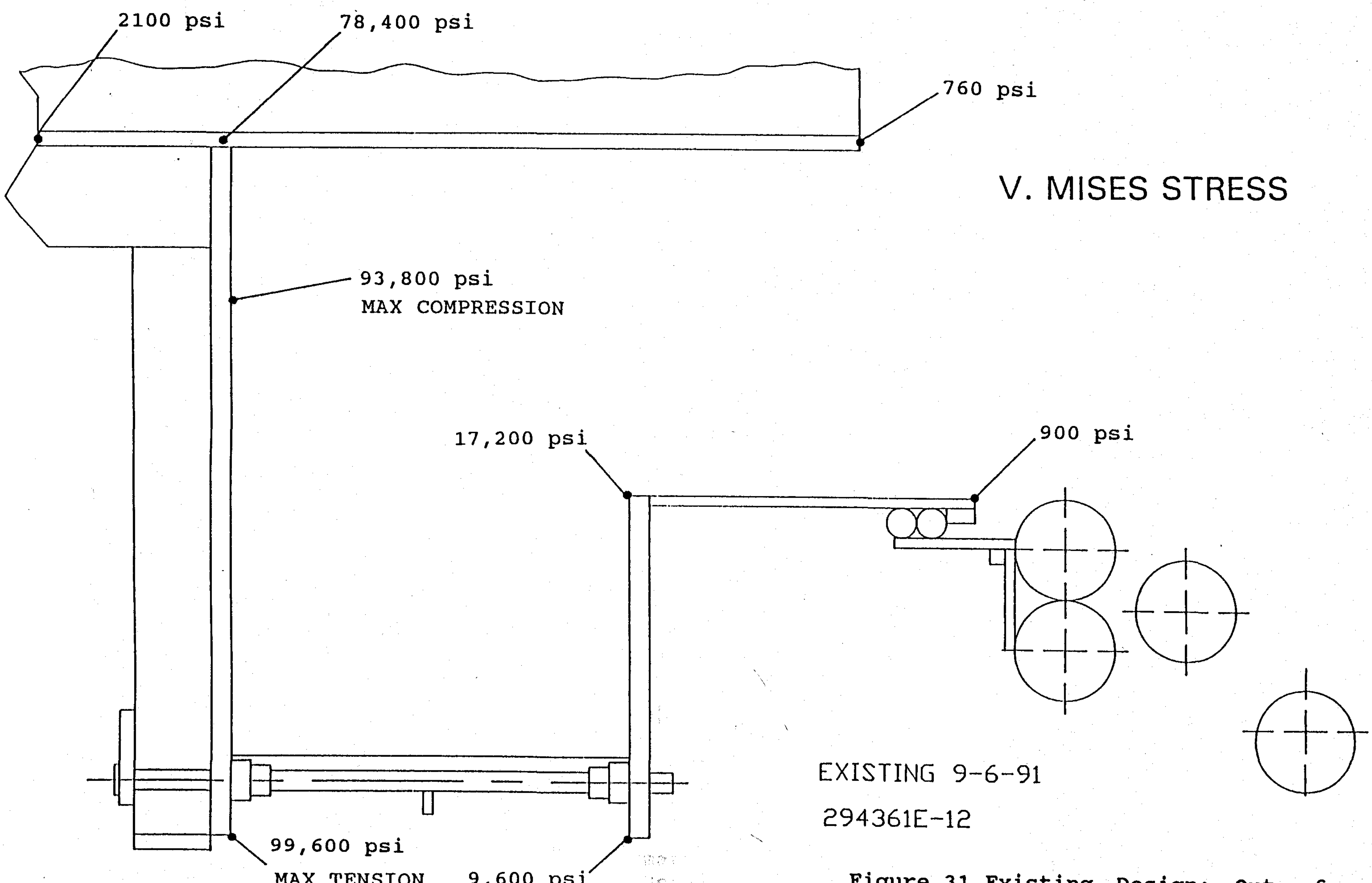


Figure 31 Existing Design: Out of Service Stress Analysis

IP7_004772

SUMMARY
EXISTING DESIGN (REF. 294361-12)
OUT OF SERVICE

BACK PLATE

- o TEMPERATURES/STRESSES AGGRAVATED BY REDUCED COOLING AIR FLOW
- o MORE SEVERE SEPARATION AND CONING/BUCKLING

INNER SLEEVE

- o MODERATE TEMPERATURES BUT STRESSES REMAIN LOW EXCEPT FOR LOCAL CONCENTRATION AT BACK PLATE ATTACHMENT
- o TEMPERATURE/STRESS RELIEF WILL OCCUR FOLLOWING SEPARATION OF BACK PLATE REDUCING LOCAL STRESS BELOW ALLOWABLE LIMITS

FRONT PLATE & THROAT SLEEVE

- o MODERATE TEMPERATURES AND STRESSES EXCEPT FOR LOCAL STRESS CONCENTRATION AT JOINT
- o PREDICTED PEAK STRESS AT JOINT APPROACHING ALLOWABLE LIMIT
- o ASSUMING SOME RECIRCULATION AND HIGHER TEMPERATURE FOR THE THROAT SLEEVE, THE STRESS WILL BE OVER THE ALLOWABLE WITH EXPECTED JOINT SEPARATION

IPPSUM2

Figure 32 Summary Existing Design: Out of Service

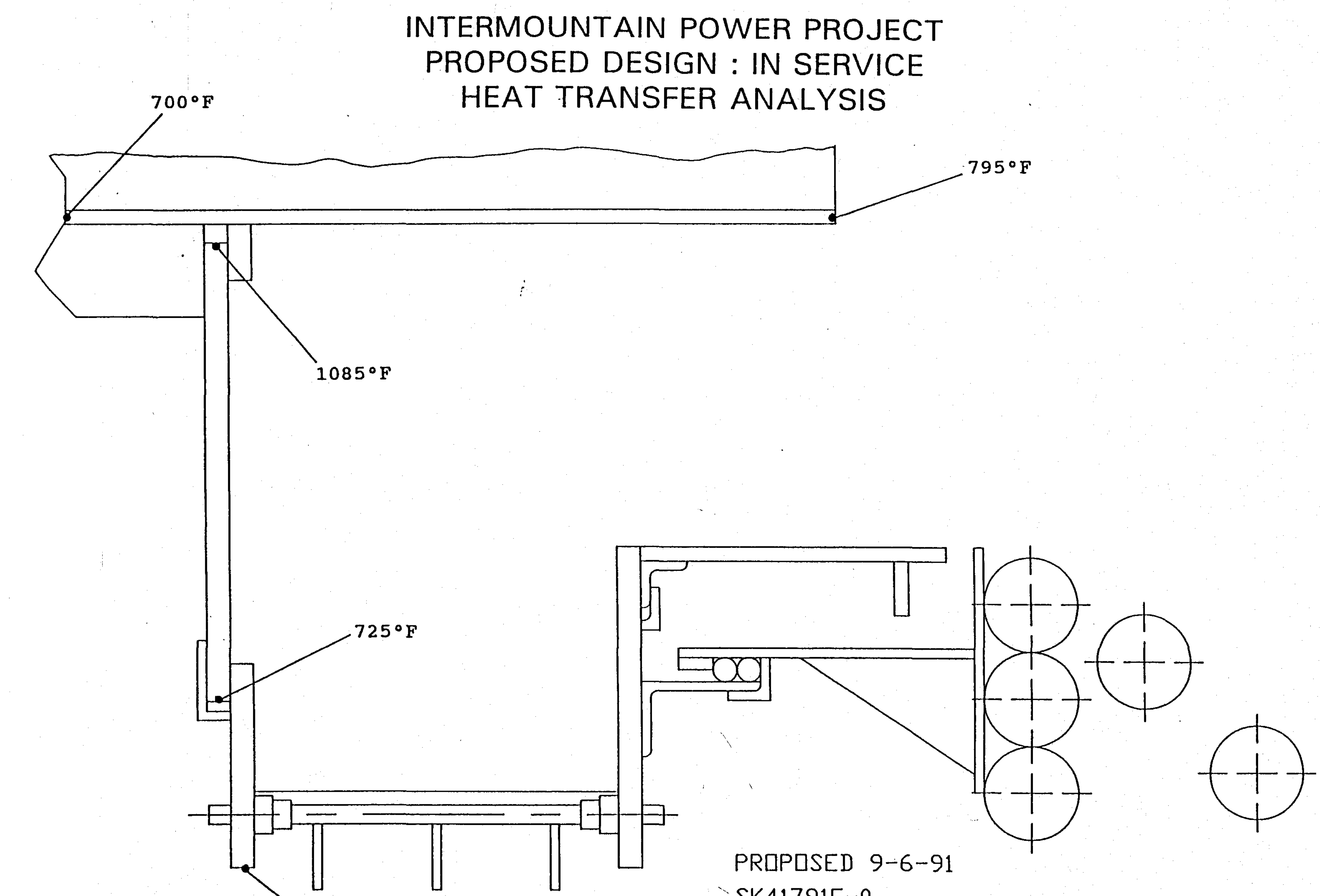


Figure 33 Proposed Design: In Service Heat Transfer

IP7_004774

INTERMOUNTAIN POWER PROJECT
PROPOSED DESIGN : IN SERVICE
STRESS ANALYSIS

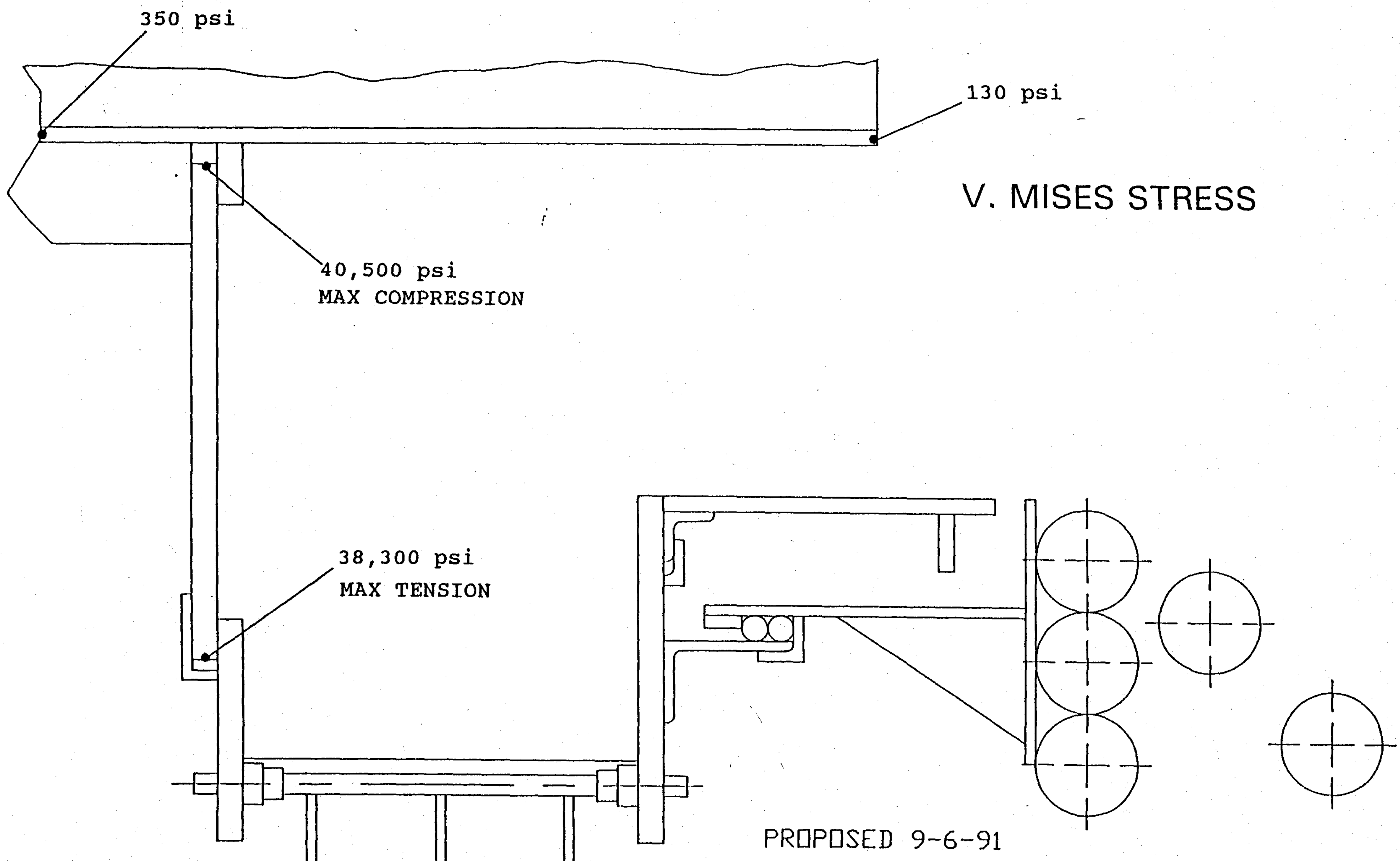


Figure 34 Proposed Design: In Service Stress Analysis

SUMMARY
PROPOSED DESIGN (REF. SK41791E-0)
IN SERVICE

BACK PLATE

- o GENERALLY SIMILAR BACK PLATE WARPING RESULTS AS IN EXISTING DESIGN
- o SLIGHTLY HIGHER TEMPERATURES DUE TO GAP AT INNER SLEEVE
- o HIGHER RADIAL TEMPERATURE GRADIENT CAUSES HIGHER TANGENTIAL STRESS GRADIENT
- o CONING/BUCKLING PREDICTED SIMILAR TO EXISTING DESIGN

INNER SLEEVE

- o LOW TEMPERATURES AND STRESSES - NO LOCAL CONCENTRATIONS

FRONT PLATE

- o ANALYSIS NOT PERFORMED - LOW TEMPERATURES/STRESSES EXPECTED THROUGHOUT

THROAT SLEEVE

- o ANALYSIS NOT PERFORMED - STRESS IN FREE CYLINDER WILL REMAIN LOW REGARDLESS OF TEMPERATURE LEVEL

IPPSUM3

Figure 35 Summary Proposed Design: In Service

INTERMOUNTAIN POWER PROJECT
PROPOSED DESIGN : OUT OF SERVICE
HEAT TRANSFER ANALYSIS

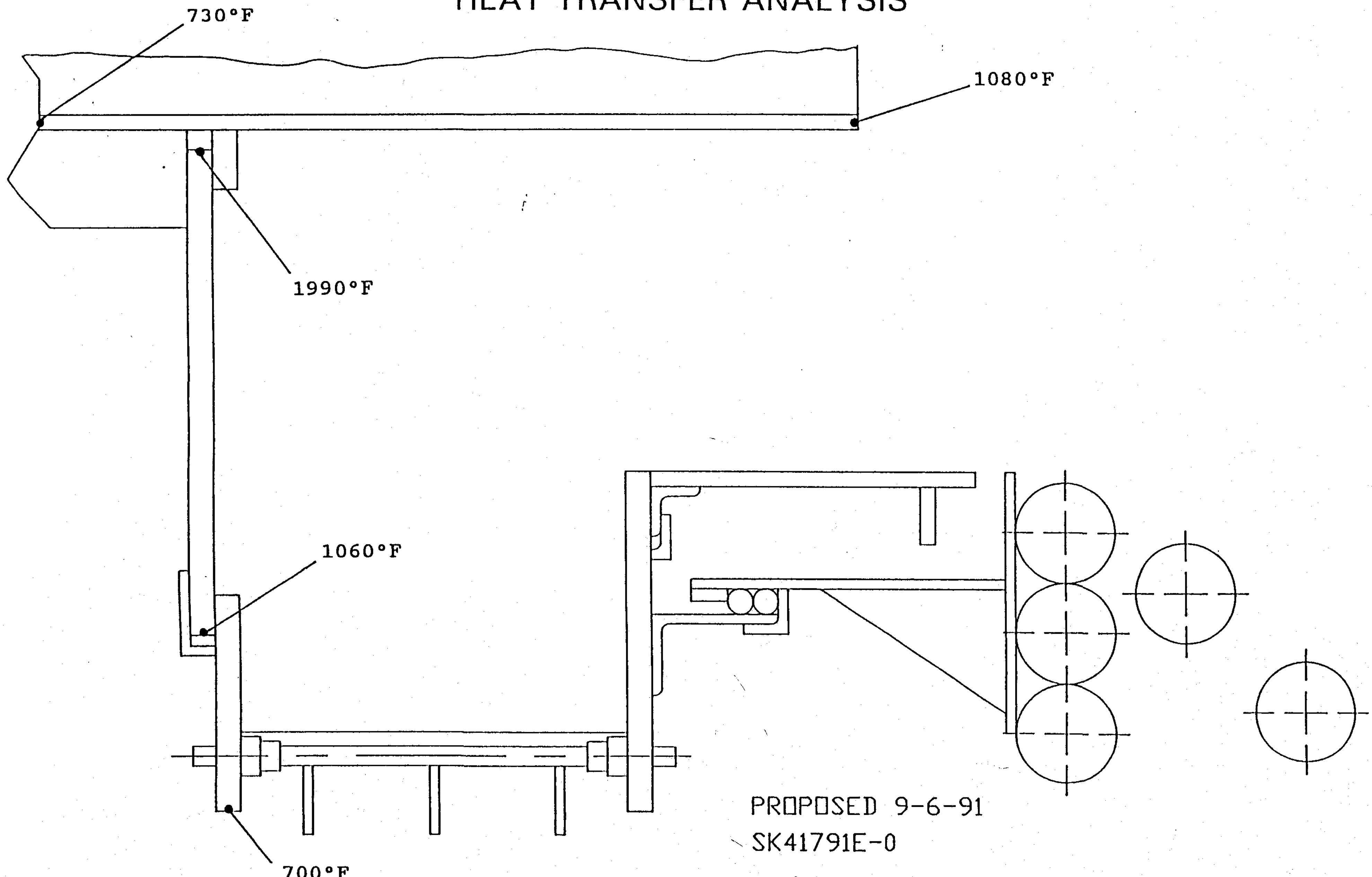


Figure 36 Proposed Design: Out of Service Heat Transfer

INTERMOUNTAIN POWER PROJECT
PROPOSED DESIGN : OUT OF SERVICE
STRESS ANALYSIS

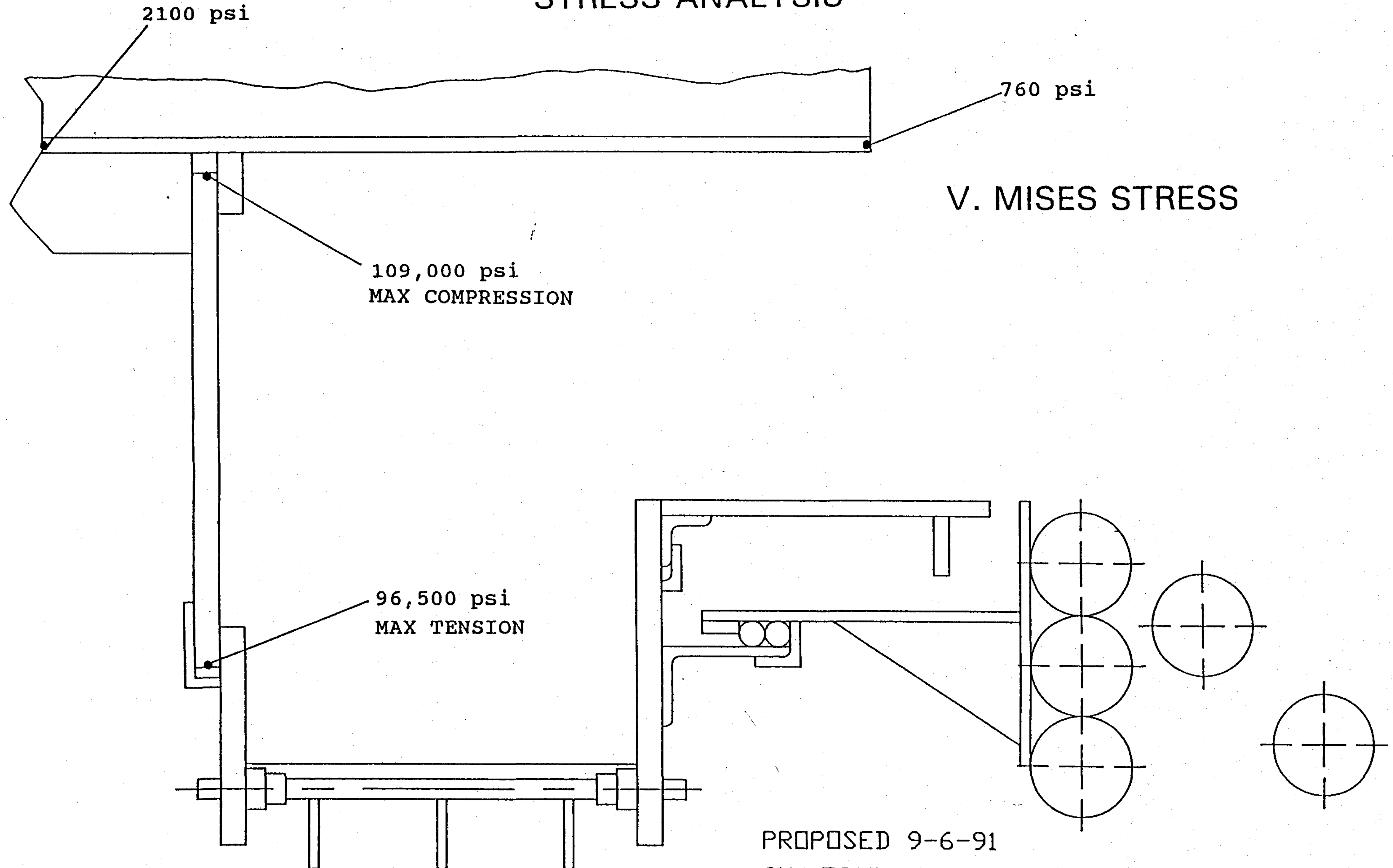


Figure 37 Proposed Design: Out of Service Stress Analysis

SUMMARY
PROPOSED DESIGN (REF. SK41791E-O)
OUT OF SERVICE

BACK PLATE AND INNER SLEEVE

- GENERALLY SIMILAR RESULTS TO "IN SERVICE", BUT HIGHER STRESSES AGGRAVATE DISTORTION.

FRONT PLATE AND THROAT SLEEVE

- ANALYSIS NOT PERFORMED - SIMILAR LOW STRESS RESULTS AS "IN SERVICE".

IPPSUM4

Figure 38 Summary Proposed Design: Out of Service

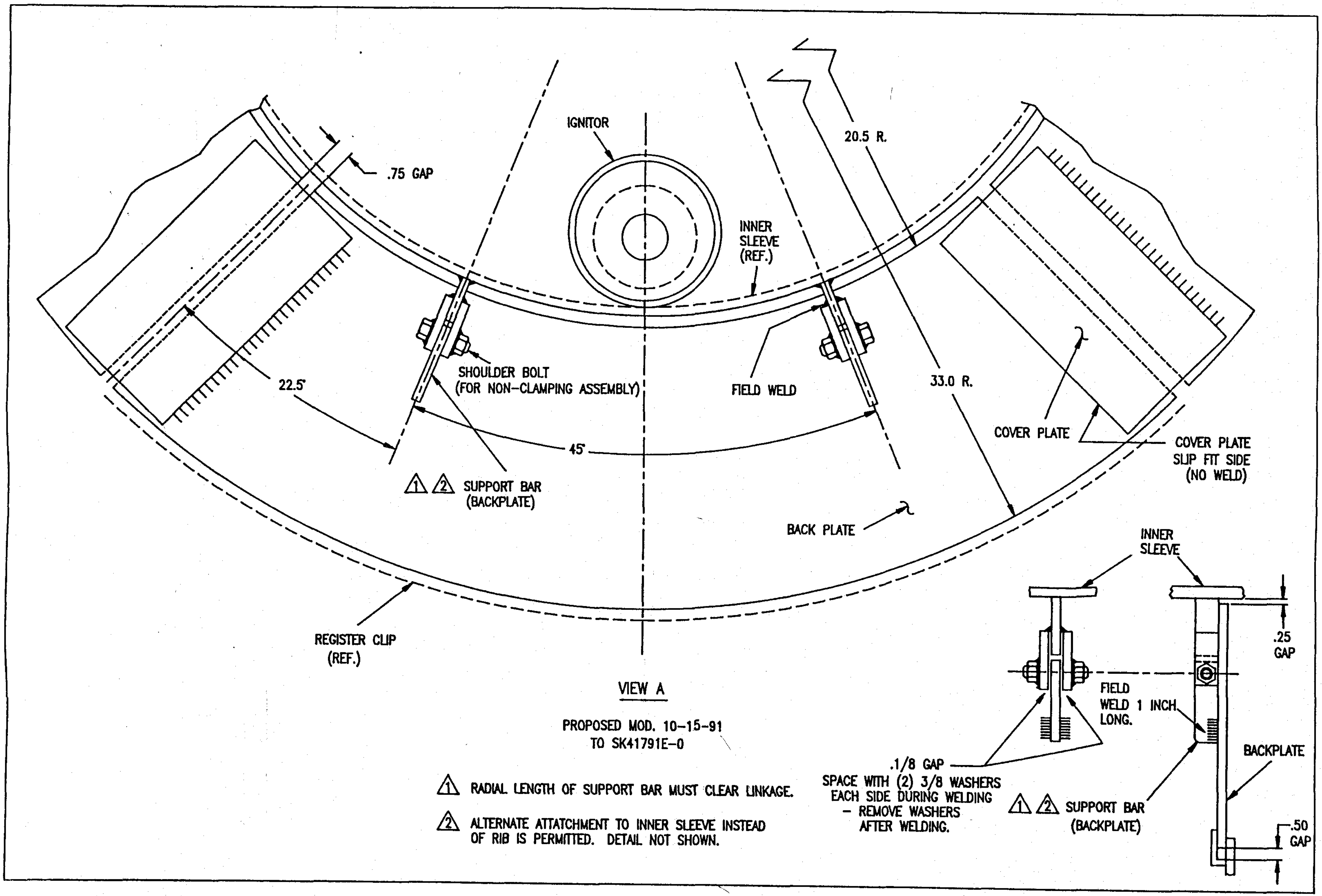


Figure 39 Segmented Back Plate

RECOMMENDED BACK PLATE DESIGN
FOUR SEGMENT PANEL: **IN SERVICE**

THERMAL GROWTH

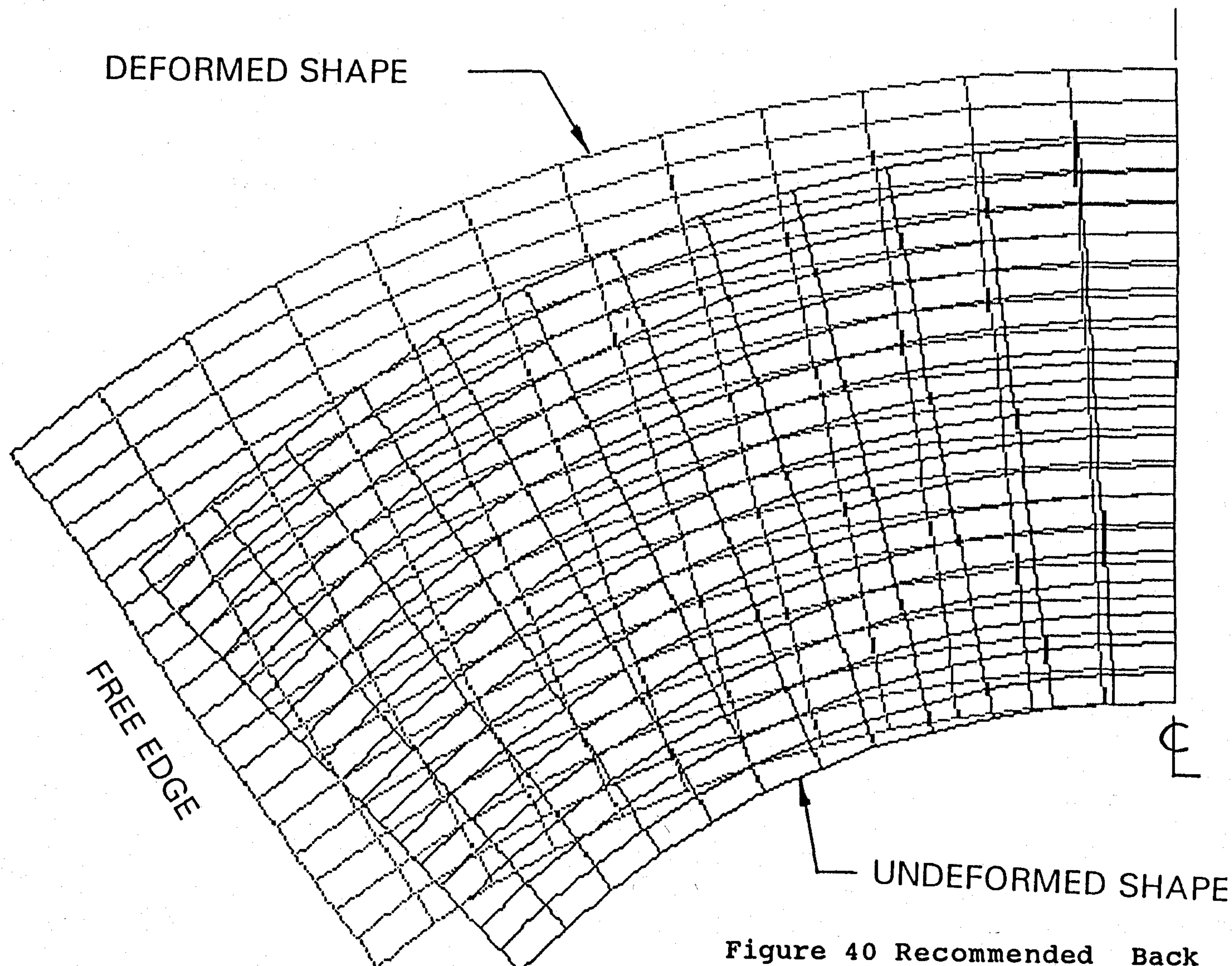
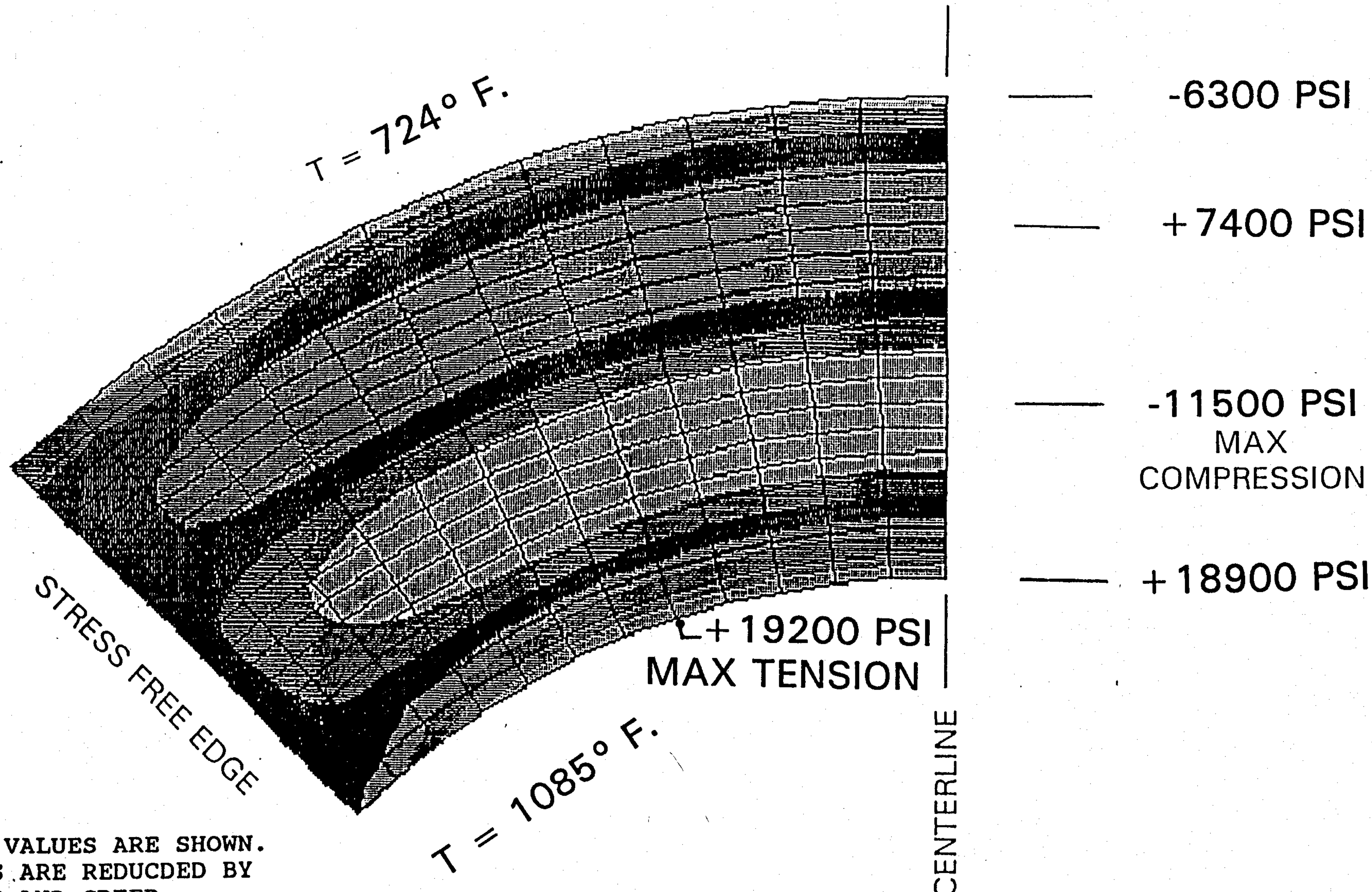


Figure 40 Recommended Back Plate Design Four Segment Panel: IN SERVICE Thermal Growth

RECOMMENDED BACK PLATE DESIGN
FOUR SEGMENT PANEL: **IN SERVICE**

TANGENTIAL STRESS



NOTE:

ELASTIC STRESS VALUES ARE SHOWN. ACTUAL STRESSES ARE REDUCDED BY PLASTIC STRAINS AND CREEP RELAXATION.

Figure 41 Recommended Back Plate Design Four Segment Panel: In Service

IP7_004782

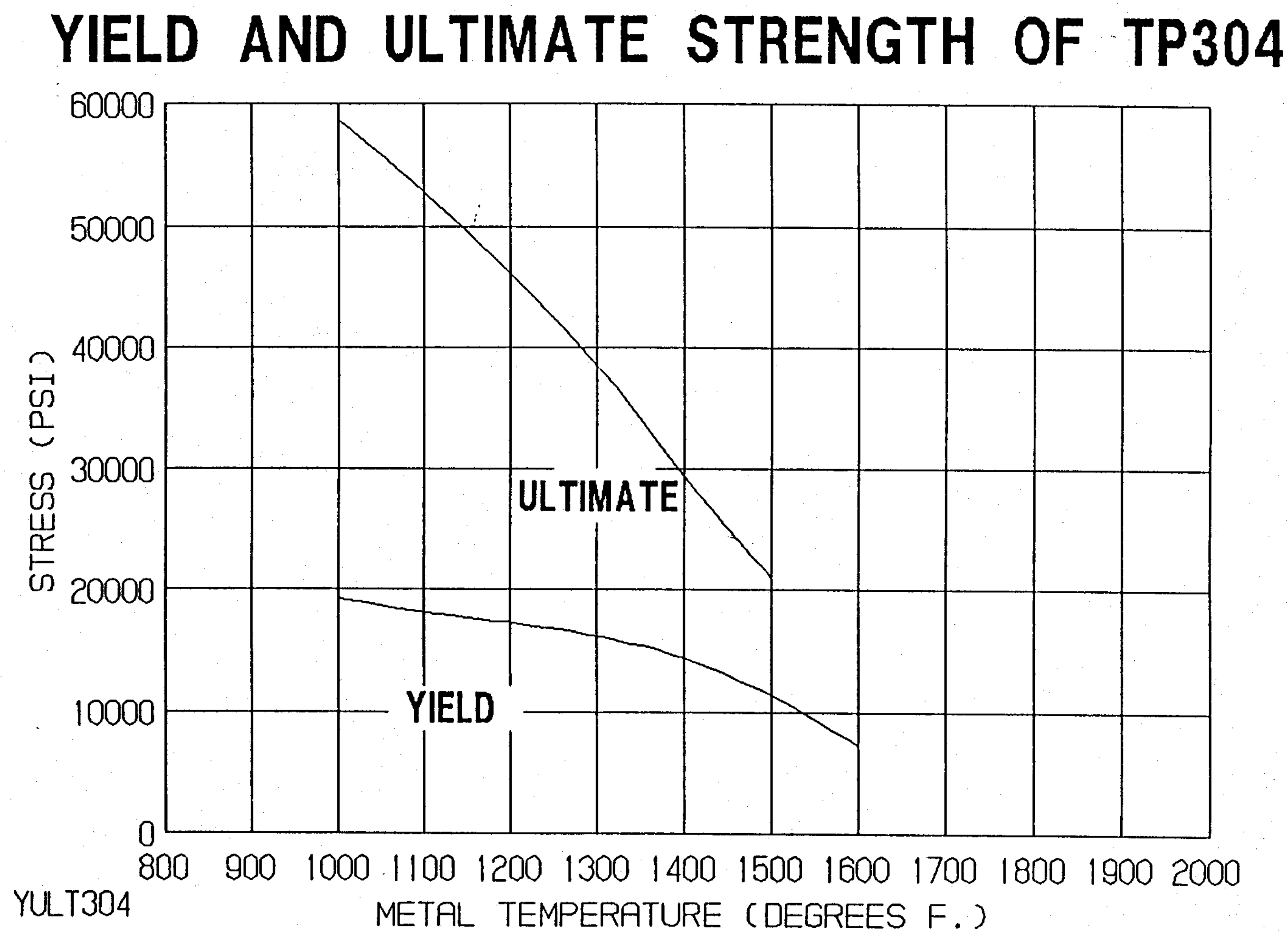
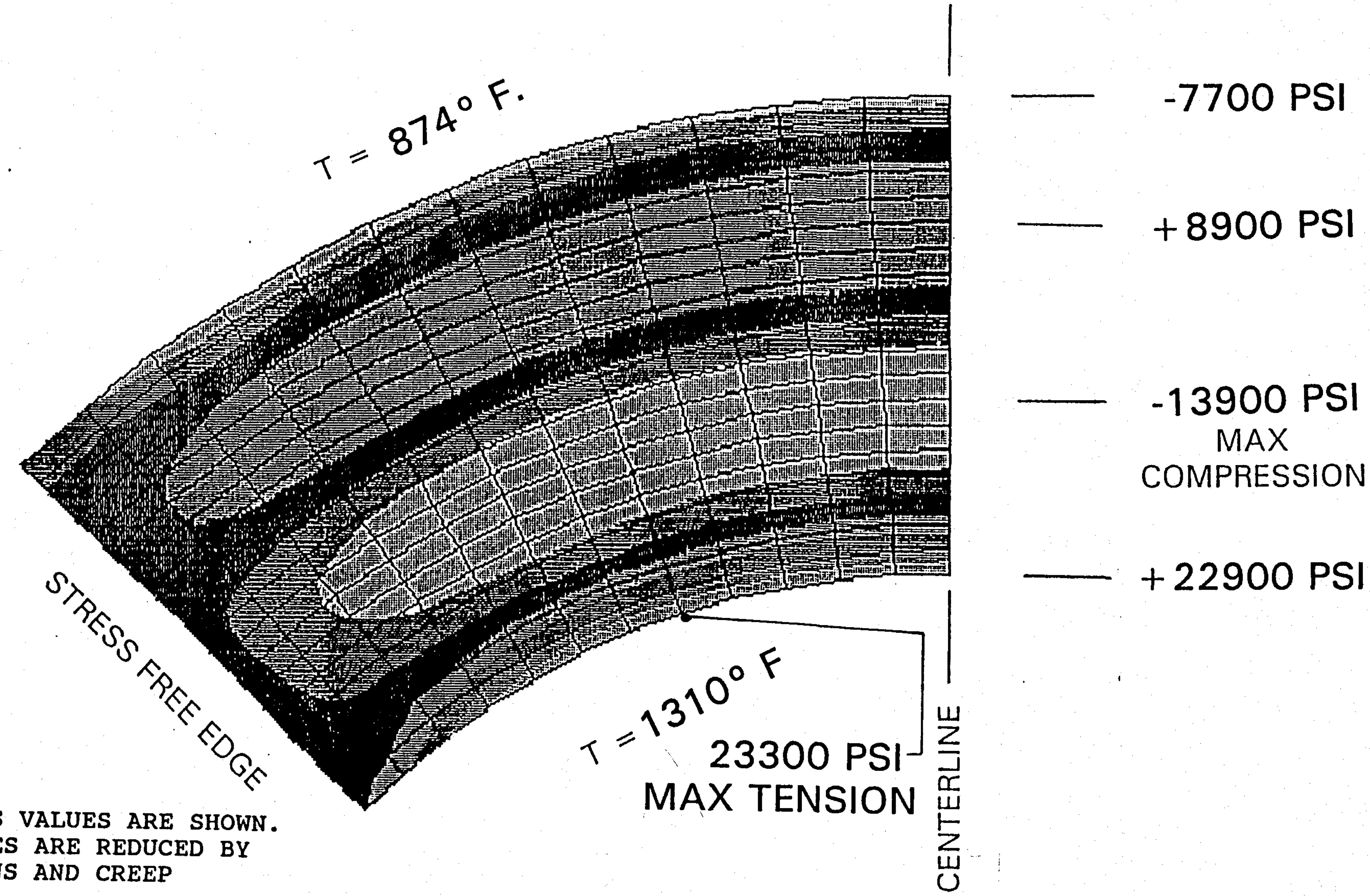


Figure 42 Yield and Ultimate Strength of TP304

RECOMMENDED BACK PLATE DESIGN
FOUR SEGMENT PANEL: OUT OF SERVICE

TANGENTIAL STRESS



NOTE:

ELASTIC STRESS VALUES ARE SHOWN. ACTUAL STRESSES ARE REDUCED BY PLASTIC STRAINS AND CREEP RELAXATION.

Figure 43 Recommended Back Plate Design Four Segment Panel: Out of Service Tangential Stress

IP7_004784

INTERMOUNTAIN POWER PROJECT
MODIFIED BACK PLATE

DESIGN

- o FOUR 90° SEGMENTED PANELS.
- o SLIP-FIT TO THE INNER SLEEVE AND OUTER REGISTER ASSEMBLY.
- o TANGENTIAL 3/4 INCH GAP BETWEEN PANELS.
- o OVERLAP PLATES BETWEEN PANELS.
- o RADIAL CENTERING BARS.

ADVANTAGES

- o ELIMINATION OF PLATE CONING/WARPING.
- o THE GAPS ALLOW FOR THERMAL GROWTH
- o OVERLAP PLATES PREVENT AIR-FLOW THROUGH GAPS.
- o RADIAL BARS TO CENTER PLATE DURING INSTALLATION AND TO PREVENT BINDING OF THE PLATE DURING THERMAL GROWTH.

IPP.MBP

Figure 44 Modified Back Plate
Design Features

INTERMOUNTAIN POWER PROJECT
CONCLUSIONS AND RECOMMENDATIONS

BACK PLATE

- o EXISTING DESIGN SEPARATION AND BUCKLING CAUSED BY HIGH TANGENTIAL STRESS GRADIENT.
- o PROPOSED DESIGN DOES NOT RELIEVE STRESS GRADIENT, SO SIMILAR SEPARATION AND BUCKLING ARE EXPECTED.
- o IT IS RECOMMENDED THAT PROPOSED SLIP FIT PLATE BE DIVIDED INTO SEPARATE PANELS TO ELIMINATE TANGENTIAL STRESS GRADIENT.
- o SAME MATERIAL AND THICKNESS AS EXISTING DESIGN IS THEREFORE ADEQUATE.

INNER SLEEVE AND THROAT SLEEVE

- o ACT AS FREE CYLINDERS WHEN SEPARATED FROM PLATES RESULTING IN LOW STRESSES REGARDLESS OF TEMPERATURE.
- o SAME MATERIAL AND THICKNESS AS EXISTING DESIGNS ARE THEREFORE ADEQUATE.

FRONT PLATE

- o EXISTING DESIGN STRESS CONCENTRATION AT JOINT WITH THROAT SLEEVE ELIMINATED BY SLIP FIT PROPOSED DESIGN.
- o SAME MATERIAL AND THICKNESS AS EXISTING DESIGN IS THEREFORE ADEQUATE.

IPPCON

Figure 45 Structural Analysis Conclusions and Recommendations

INTERMOUNTAIN POWER PROJECT
RECOMMENDED MATERIALS
(BASED ON SLIP FIT/SEGMENTED PANEL BACK PLATE AND SLIP FIT FRONT PLATE)

COMPONENT	MATERIAL
REGISTER FRONT PLATE	1/2" A36 PLATE *
REGISTER BACK PLATE	1/2" TP304H PLATE *
INNER AIR SLEEVE	1/4" TP309H PLATE *
THROAT SLEEVE	1/4" TP304H PLATE *
SLIP SEAL CASING	3/16" TP304H PLATE *
COAL PIPE TIP	1/2" TP309H PLATE *

(* - DENOTES MATERIAL AS IS CURRENTLY INSTALLED)

IPP.RM

Figure 46 Recommended Materials

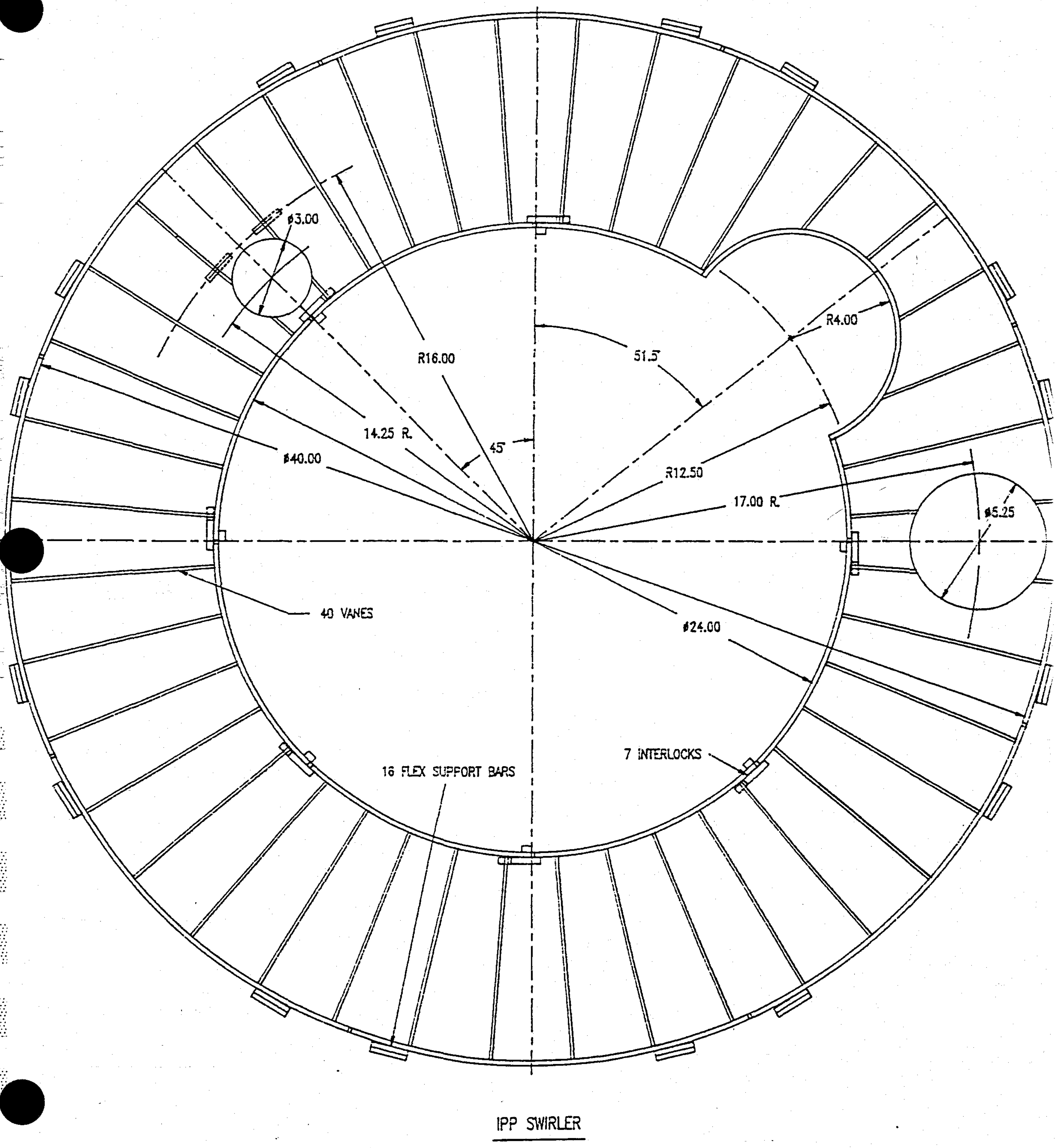


Figure 47 Intermountain Power Project Swirler

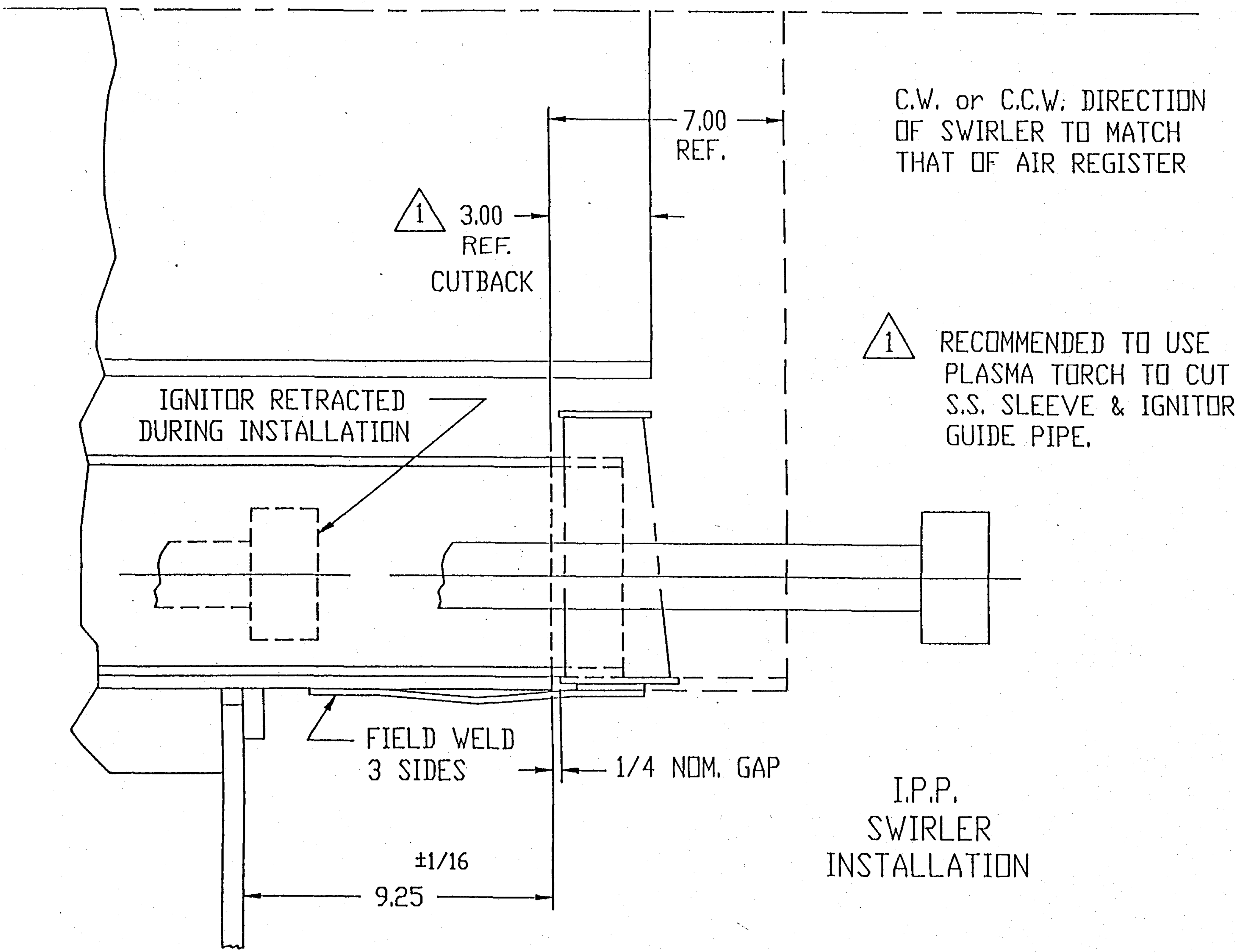


Figure 48 Swirler Installation

IP7_004789

INTERMOUNTAIN POWER PROJECT
SWIRLER

DESIGN

- o 40 VANES WELDED TO INNER AND OUTER SHROUD
- o ATTACHES TO COAL NOZZLE BY 16 FLEX BAR SUPPORTS
- o INNER SHROUD INTERLOCK PINNED TO SEGMENTS

ADVANTAGES

- o SEGMENTED DESIGN ALLOWS FOR THERMAL GROWTH BETWEEN THE OUTER SHROUD AND THE COAL NOZZLE
- o INTERLOCK PIN DESIGN PERMITS RADIAL AND TANGENTIAL THERMAL GROWTH WHILE CONSTRAINING AXIAL SEGMENT MOVEMENT

IPP.SWL

Figure 49 Swirler Design Features

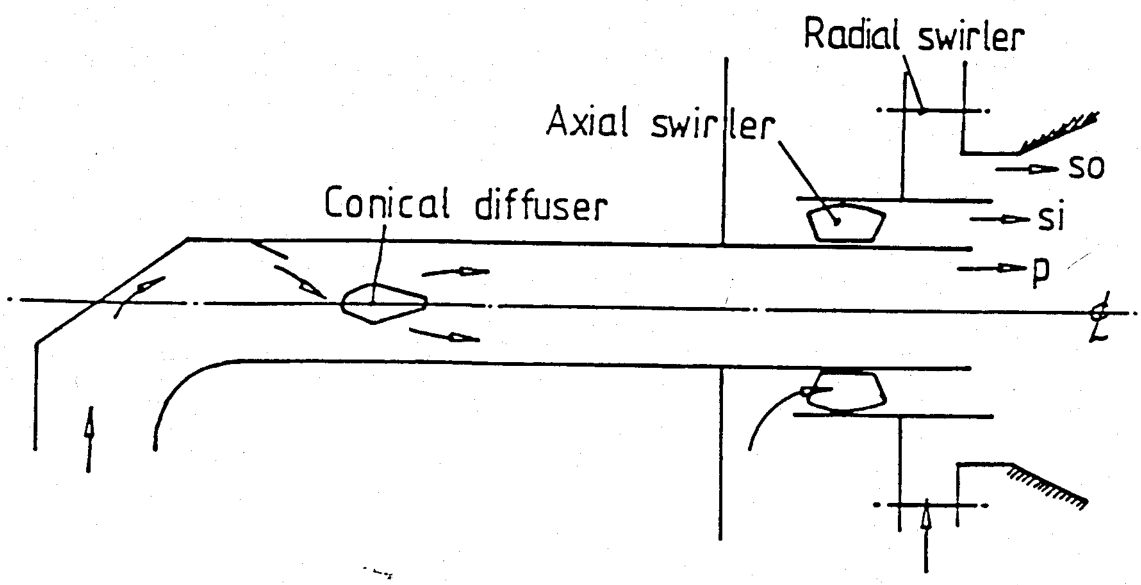


Figure 50 A dual-register burner. (After La Rue (1982).)

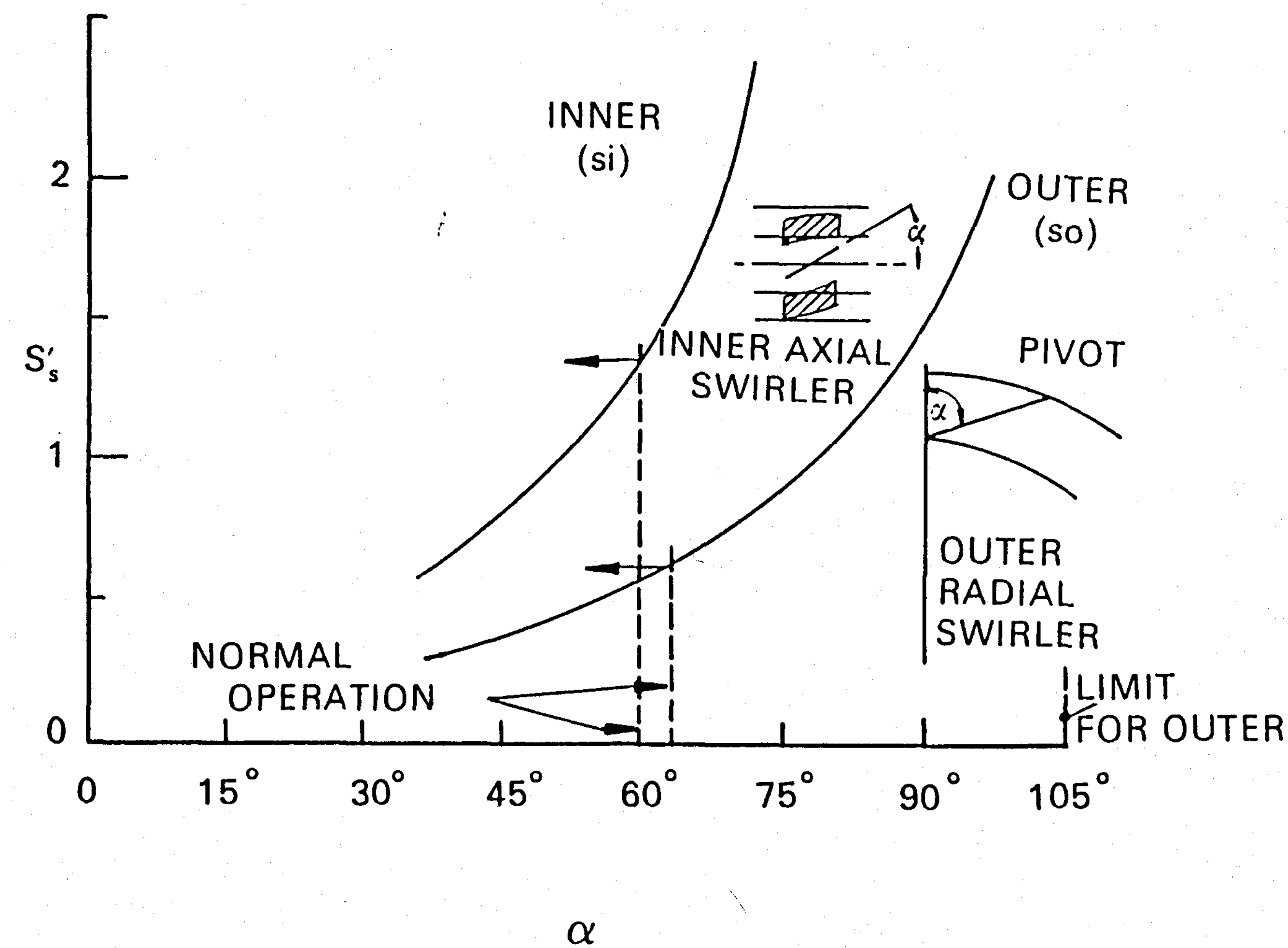


Figure 51 Swirl numbers estimated for the flows from the two registers of Figure 50

IP7_004792

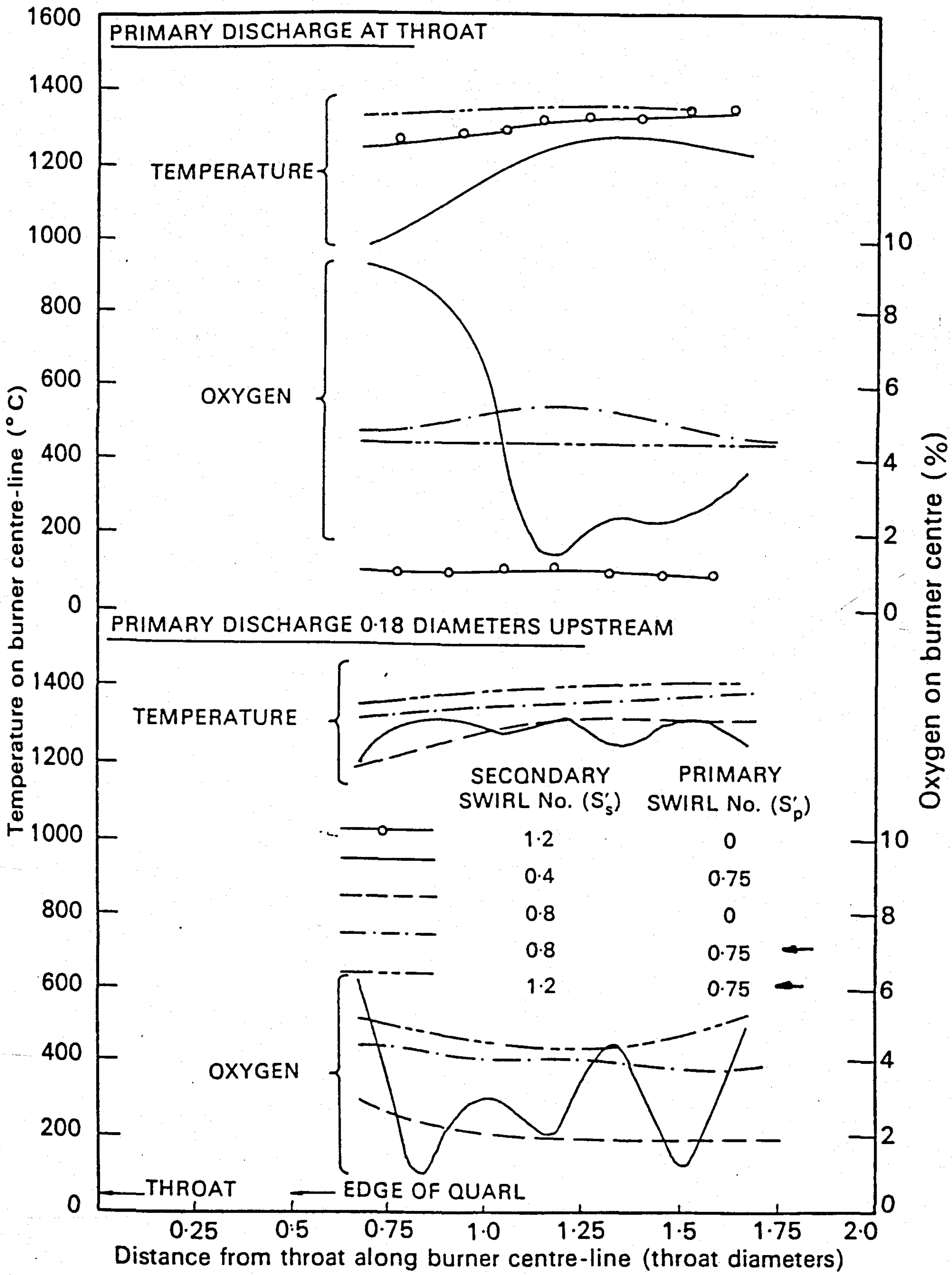


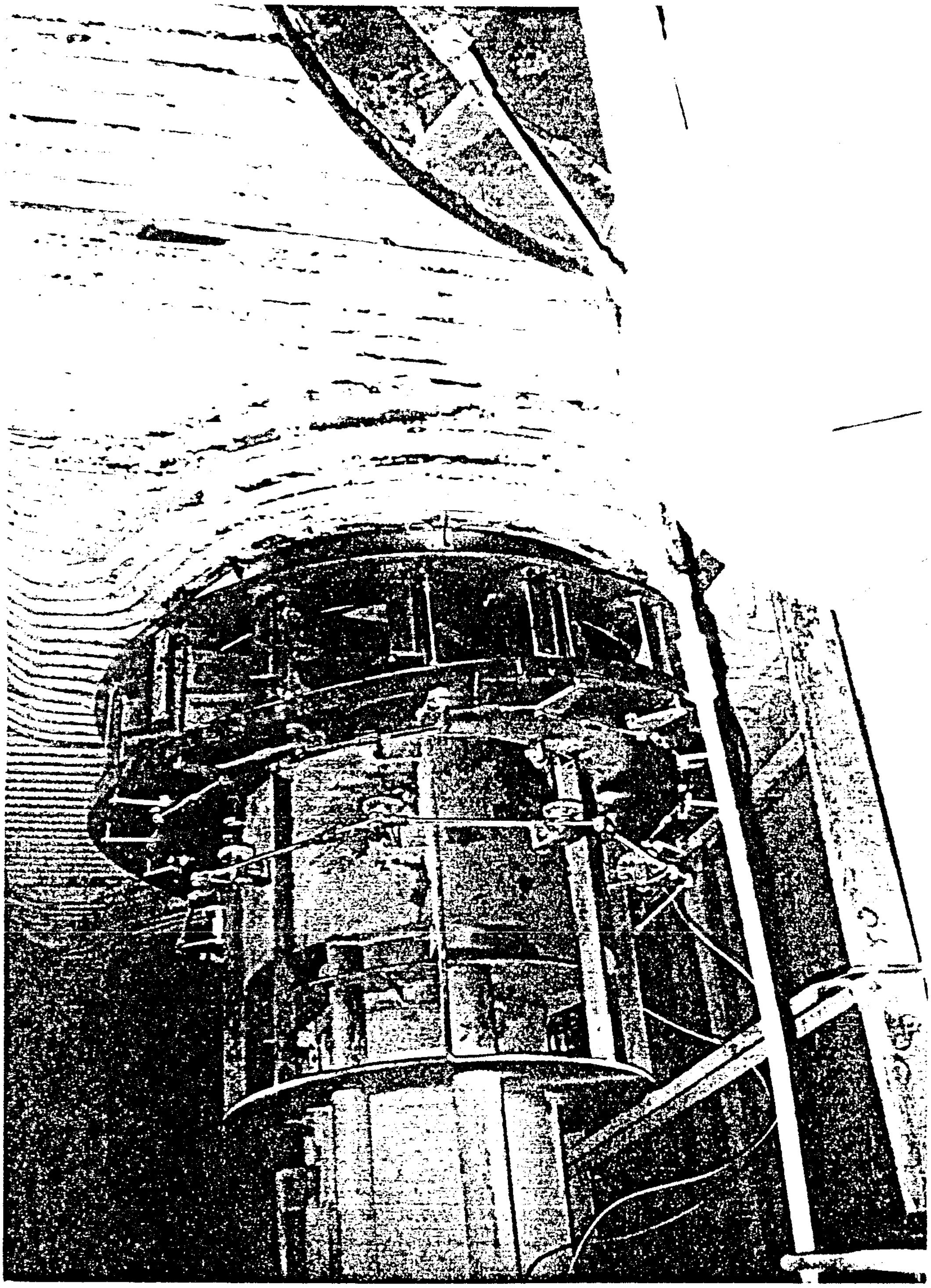
Figure 52 The effect of swirl on axial temperature and oxygen profiles. (After Jeremiejczyk (1978).)

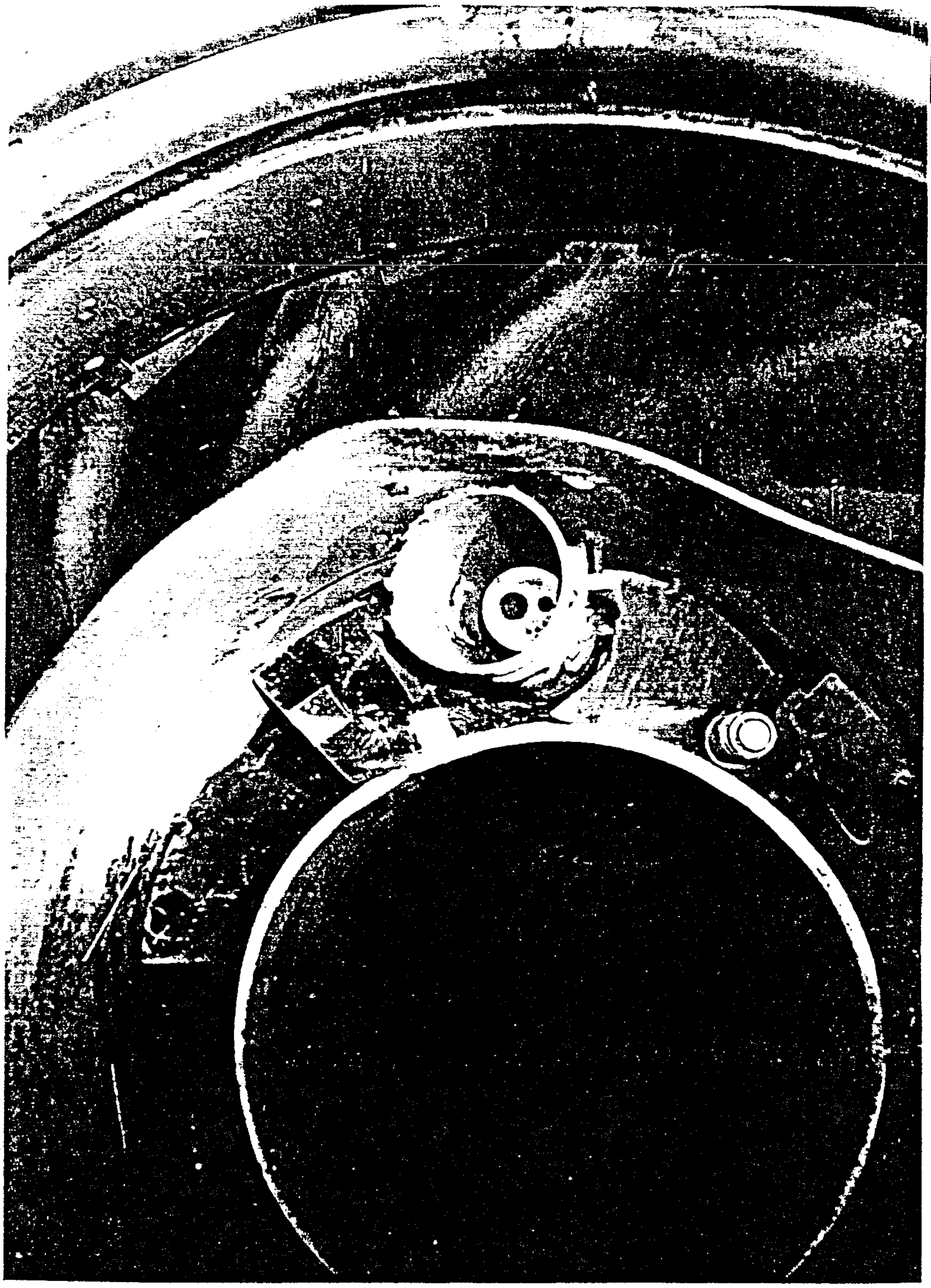
IP7_004793

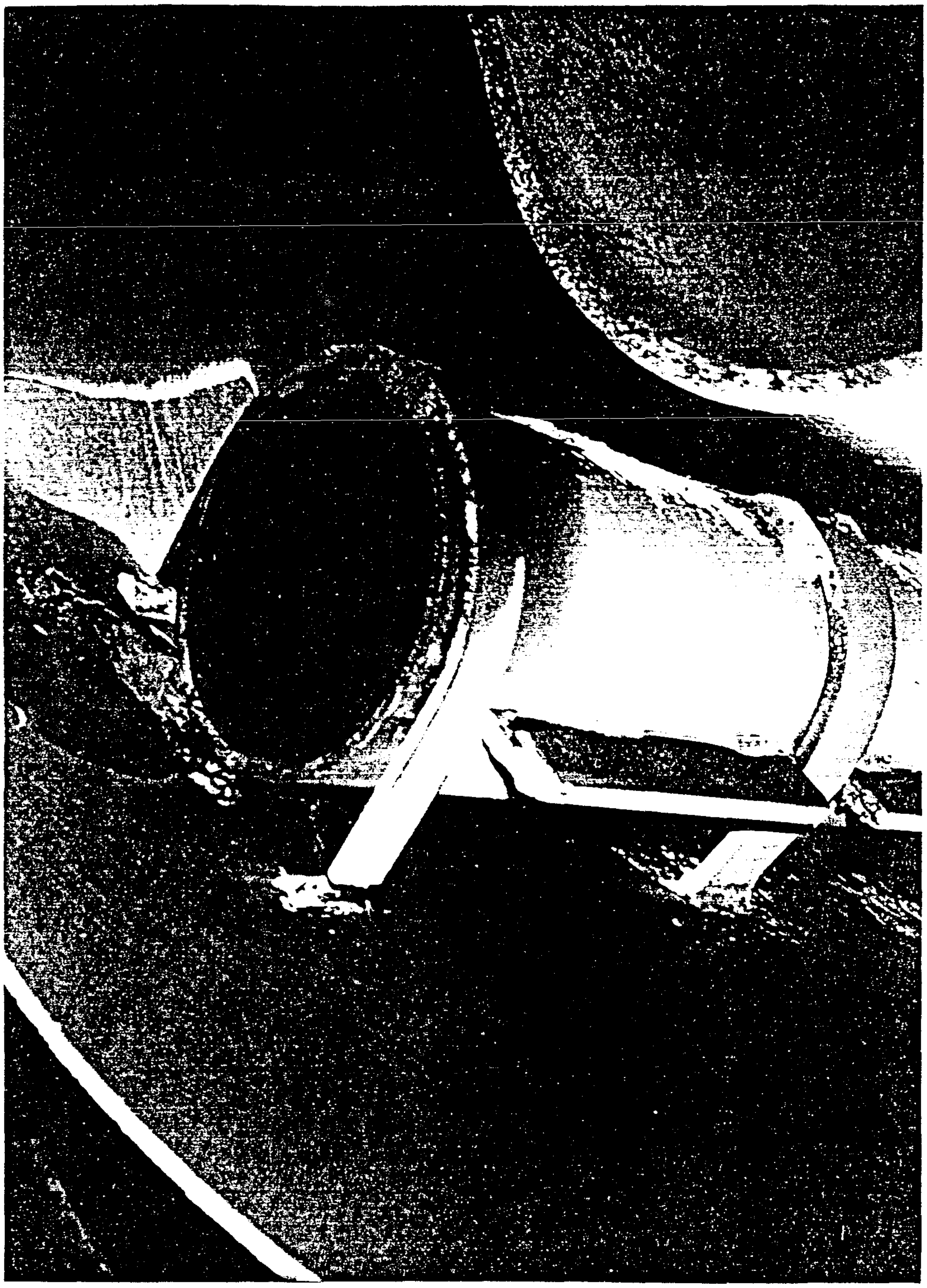
APPENDIX I

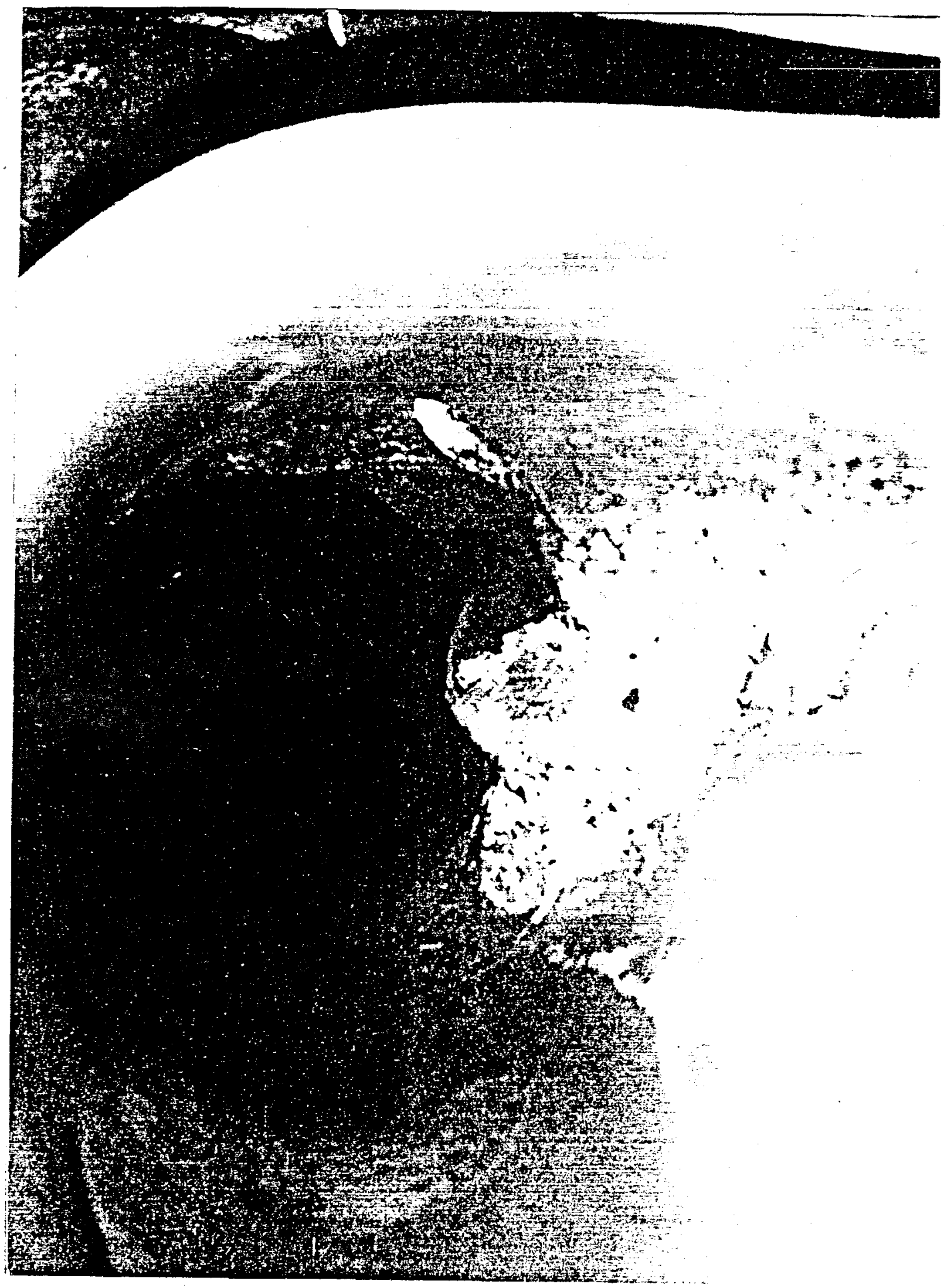
INTERMOUNTAIN POWER PROJECT

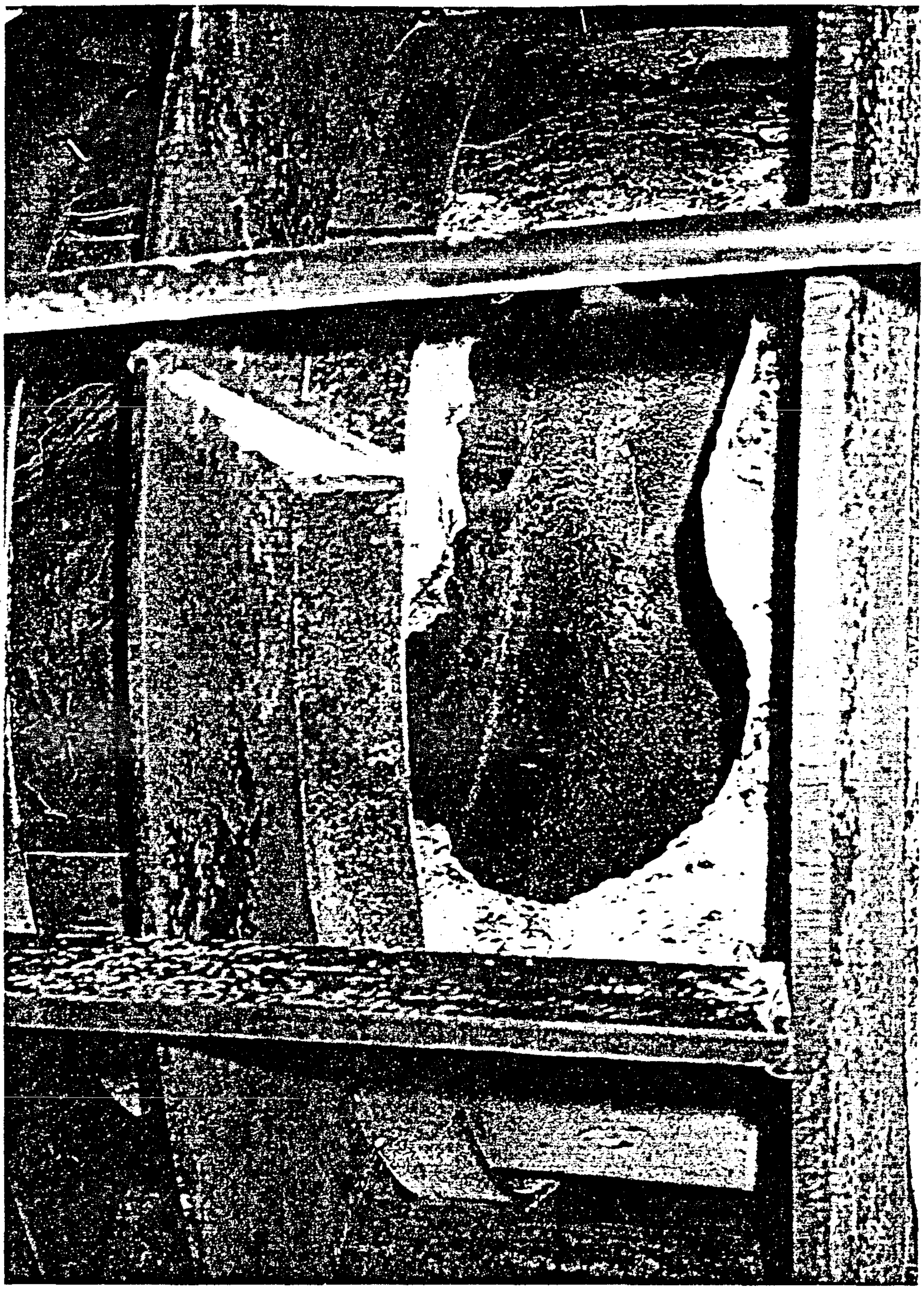
BURNER PHOTOGRAPHS

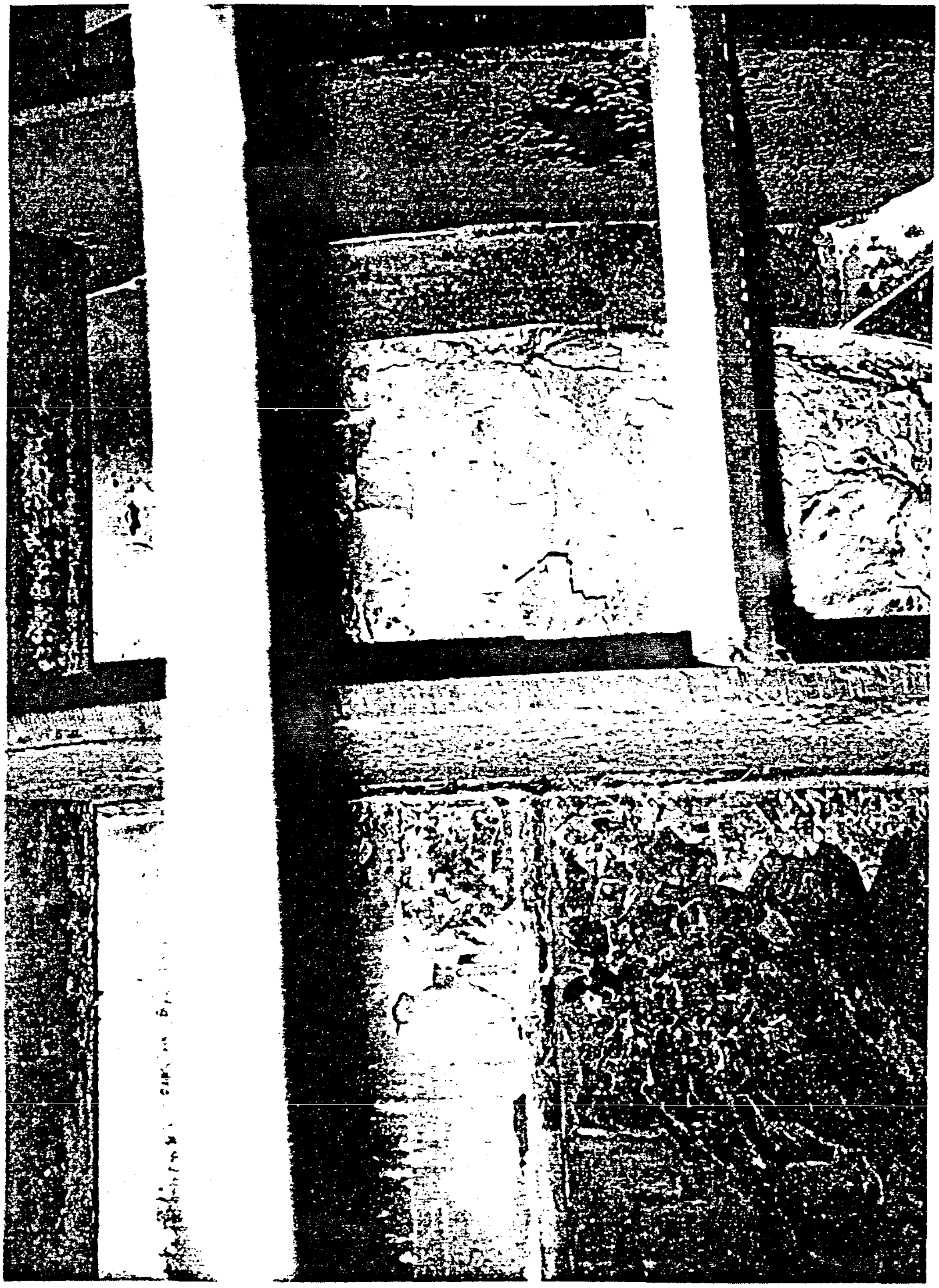


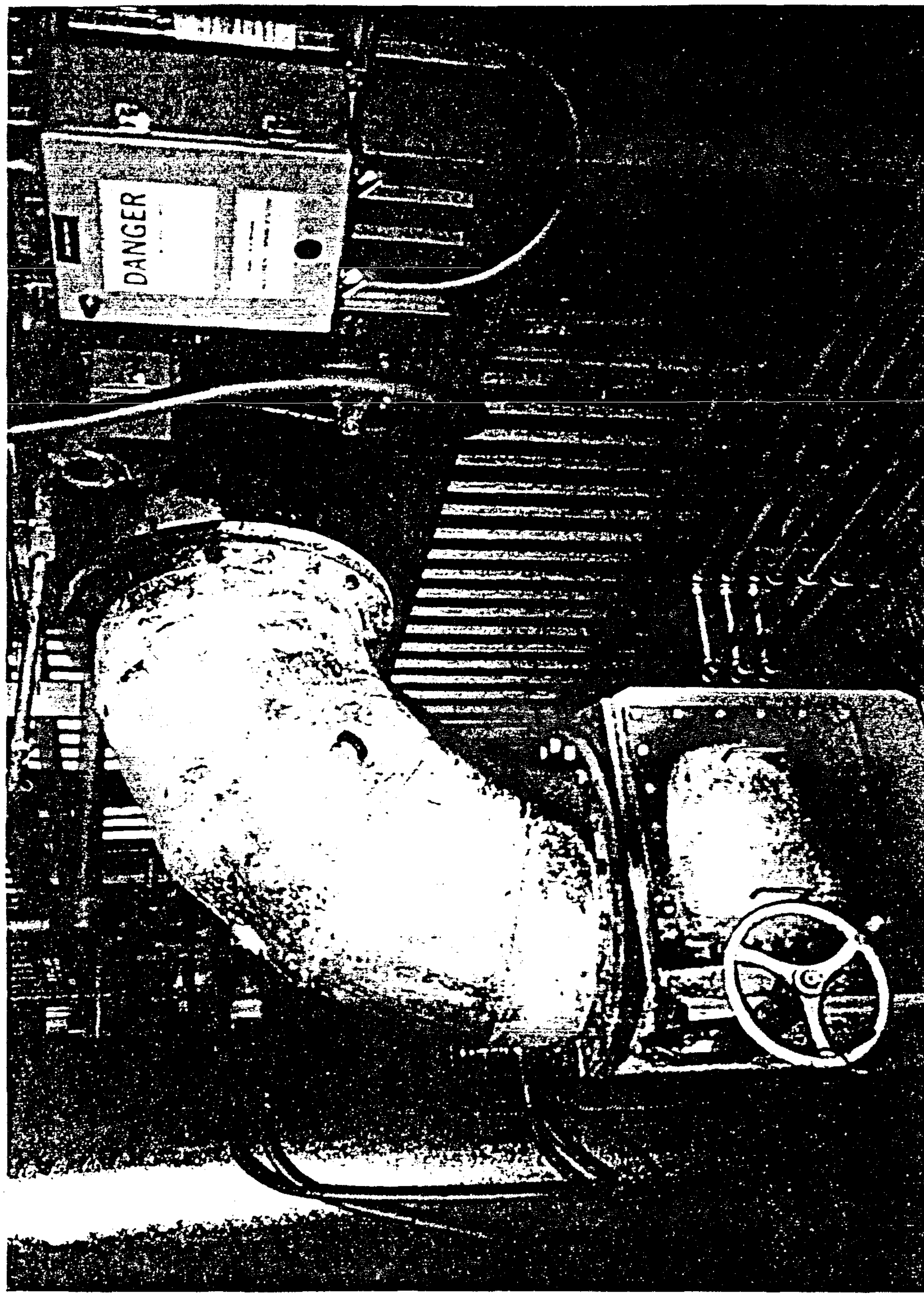


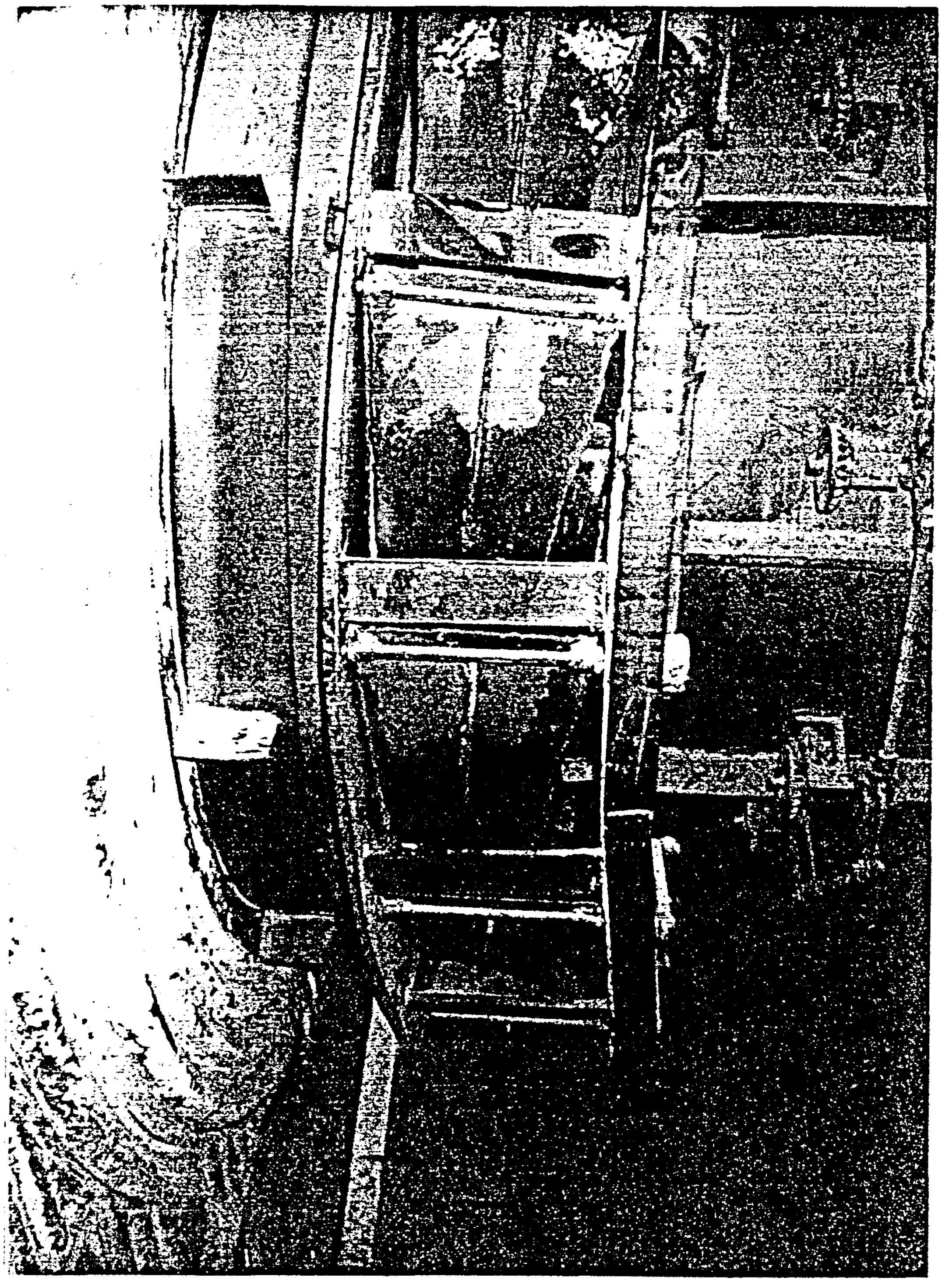


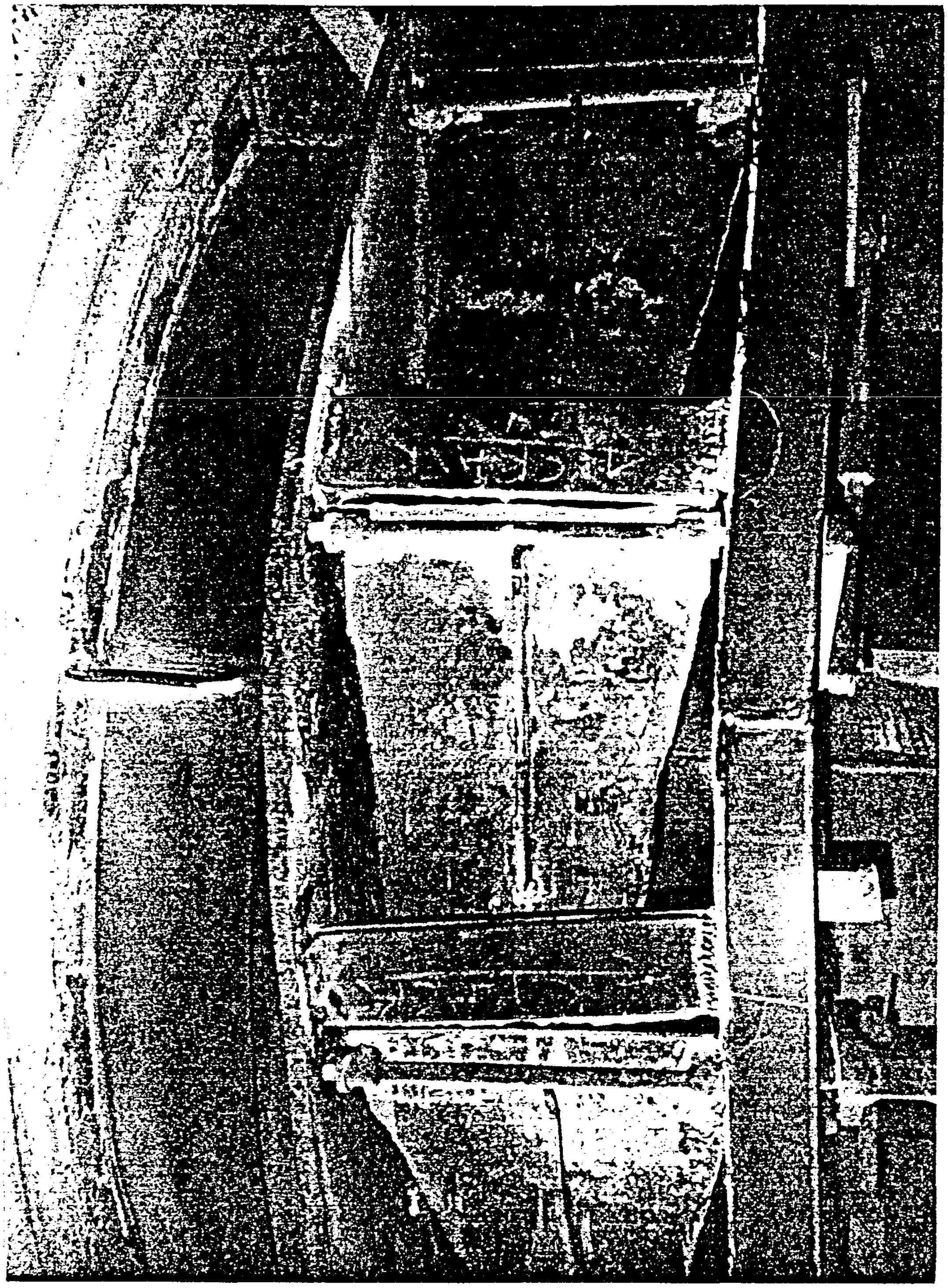


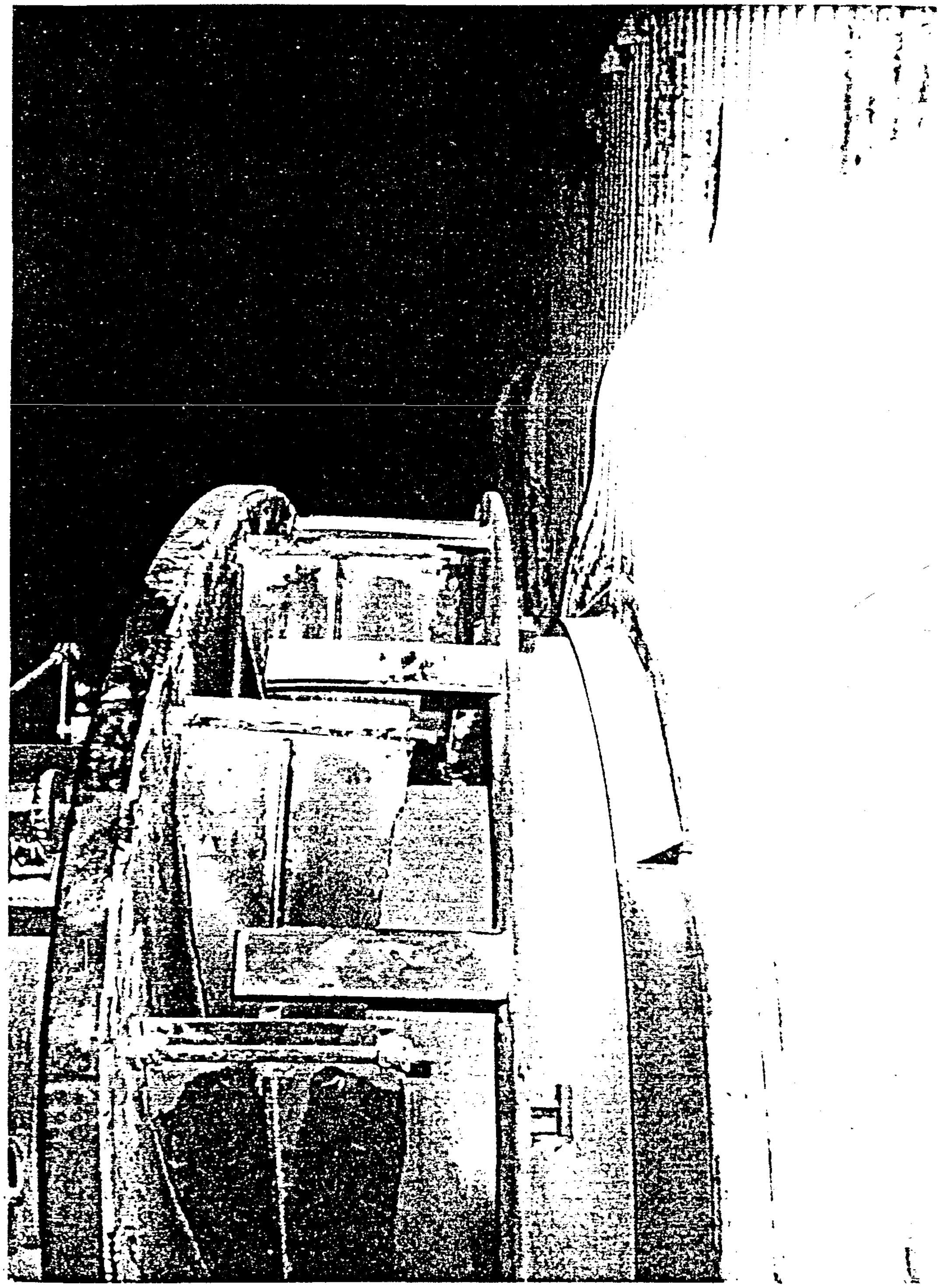


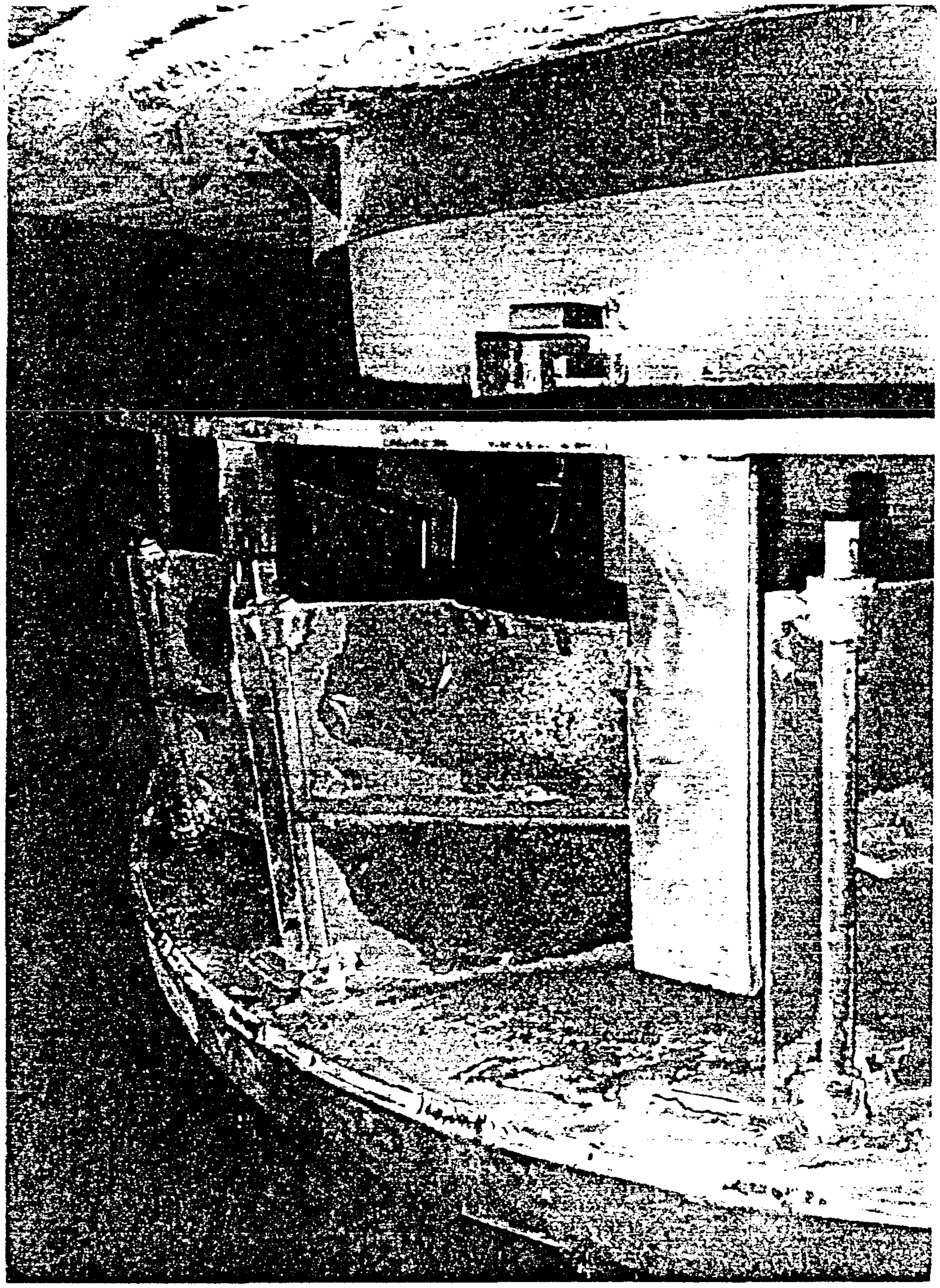


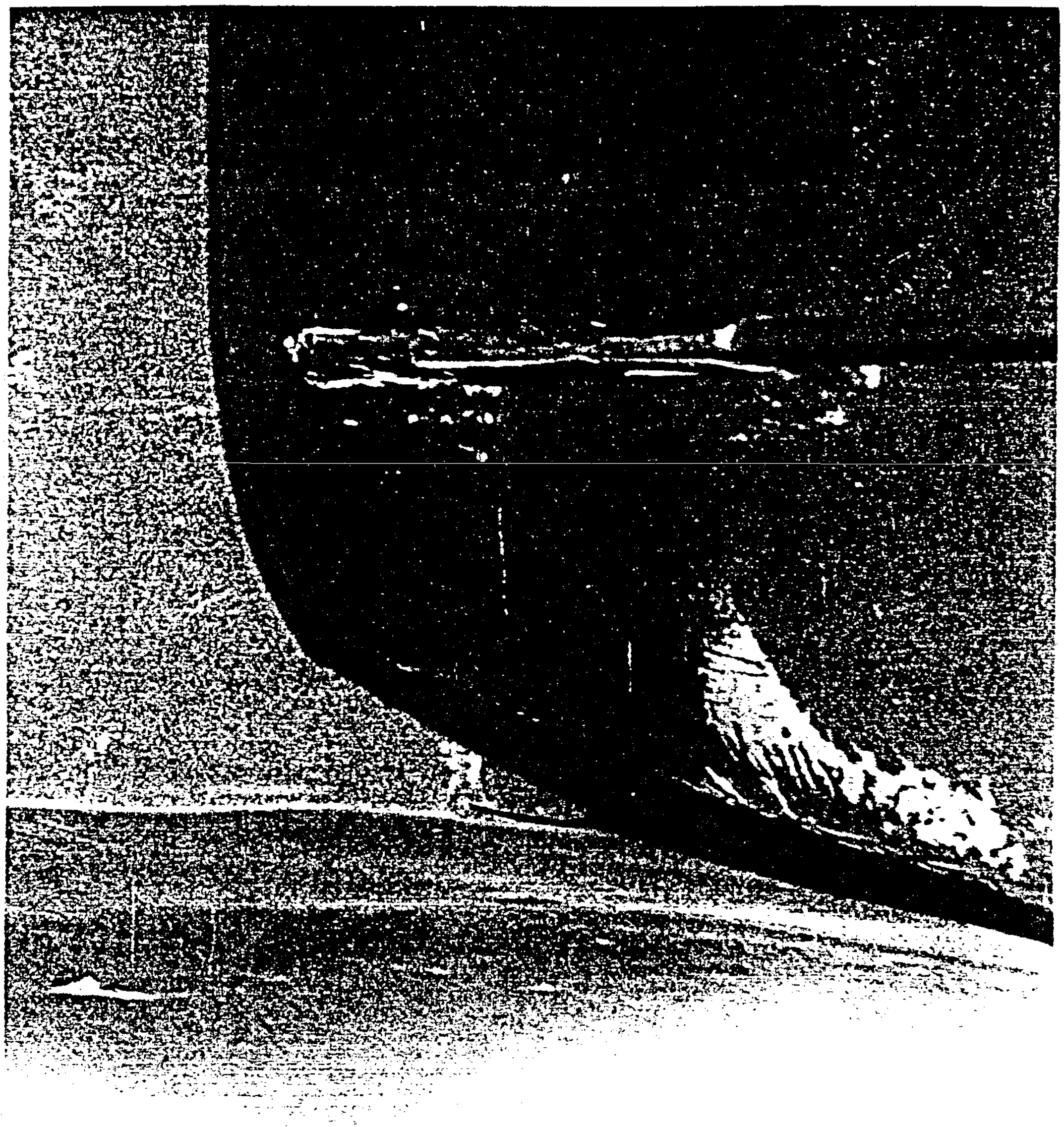

DANGER

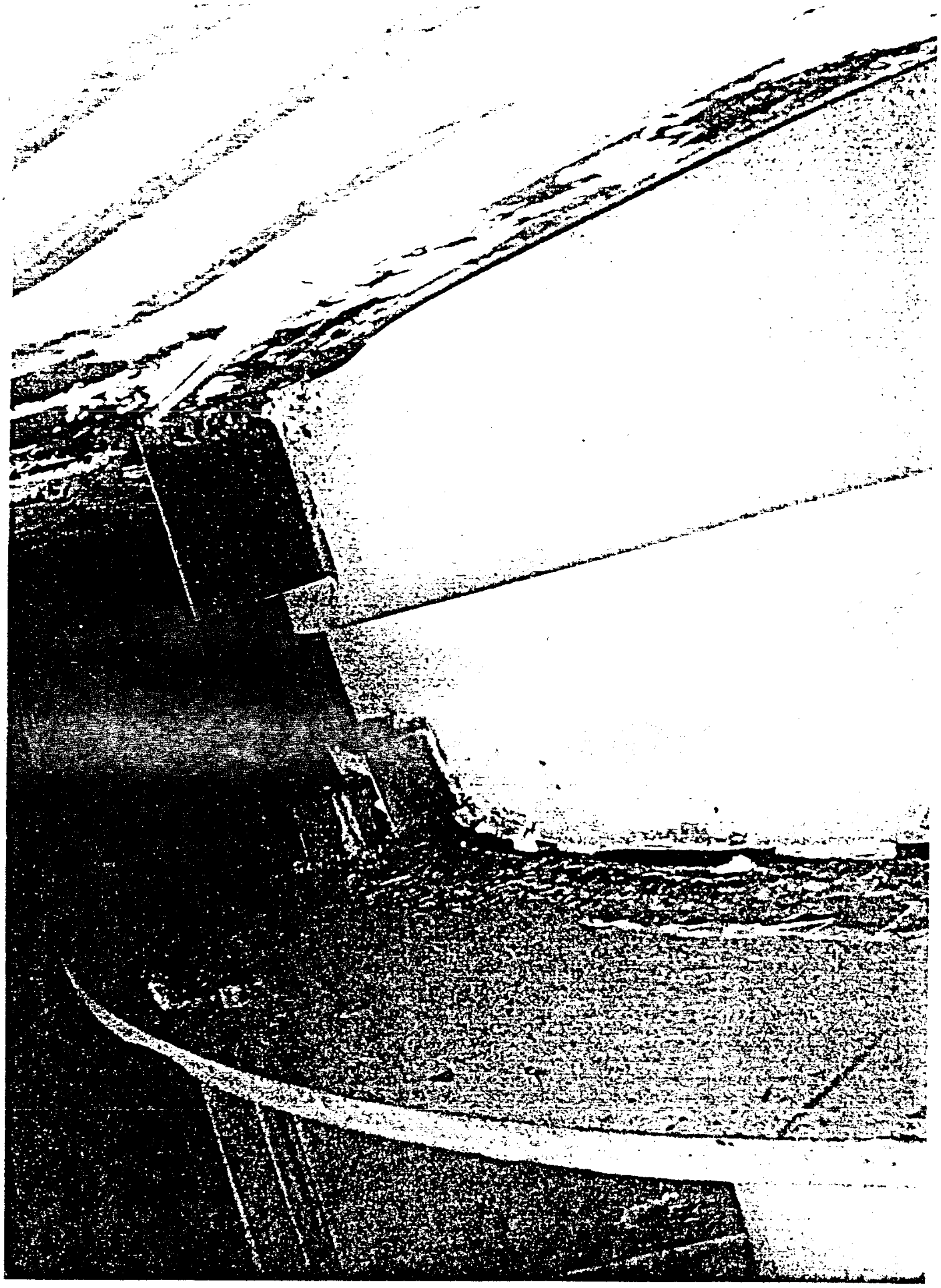


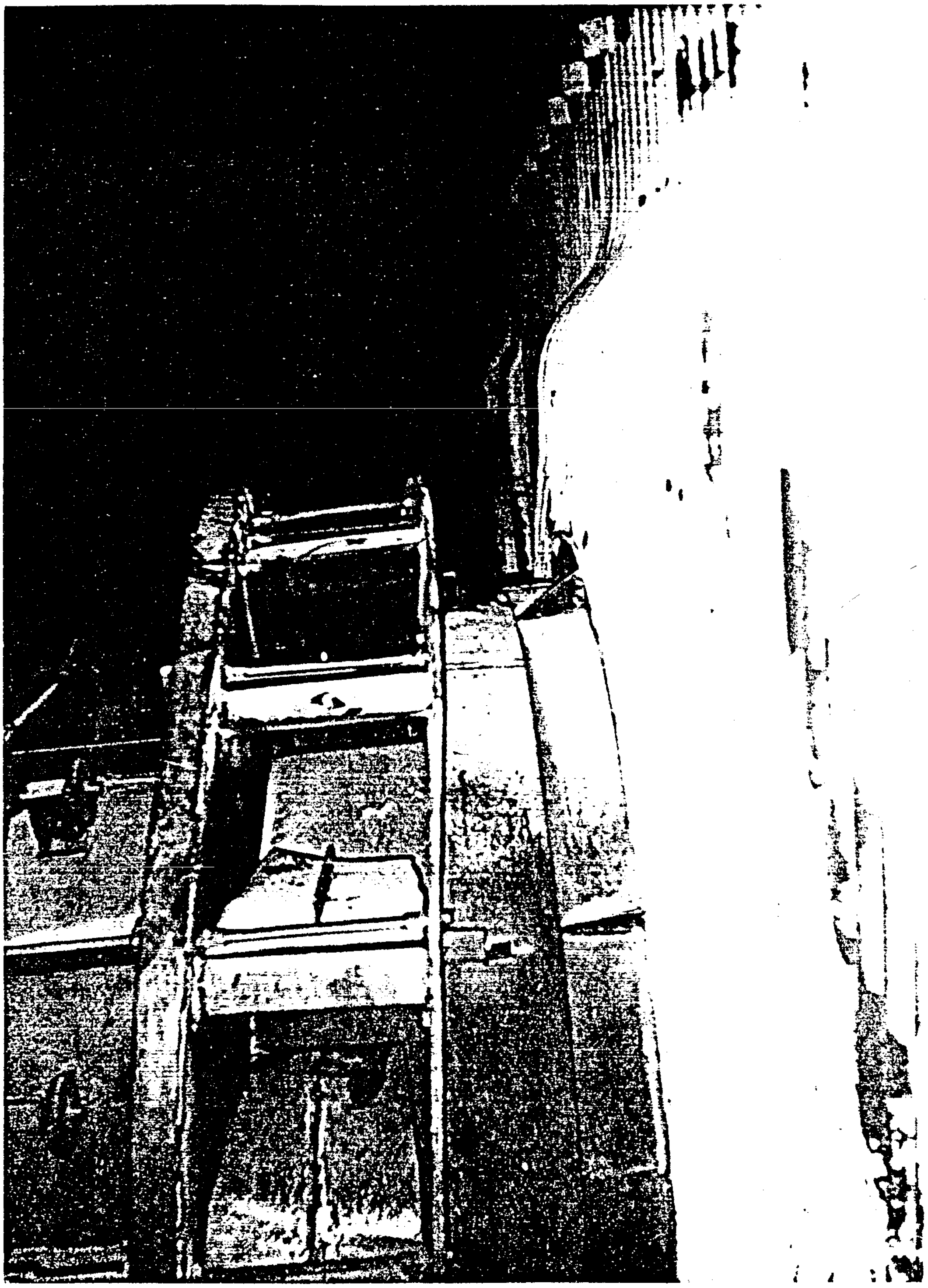


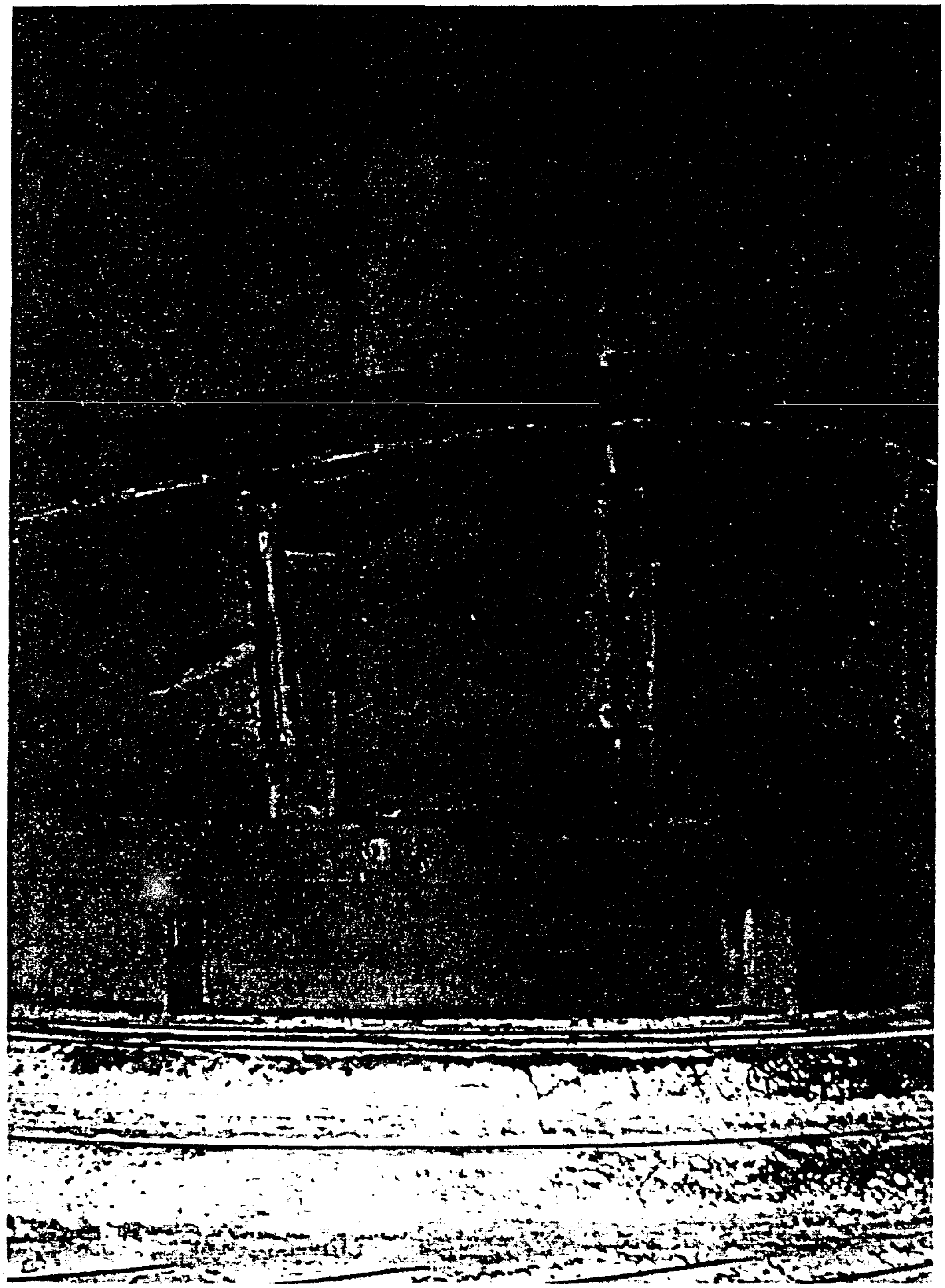


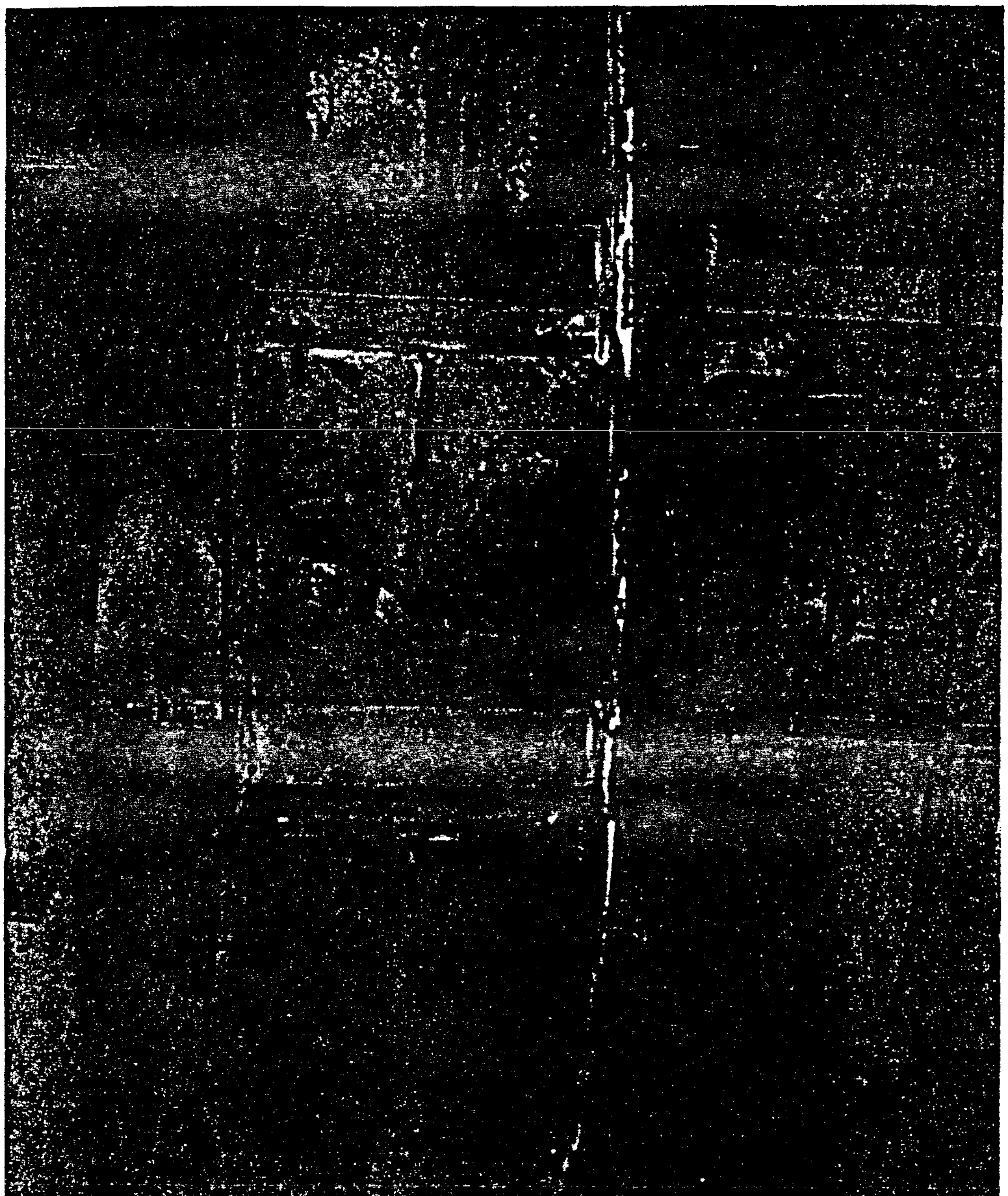


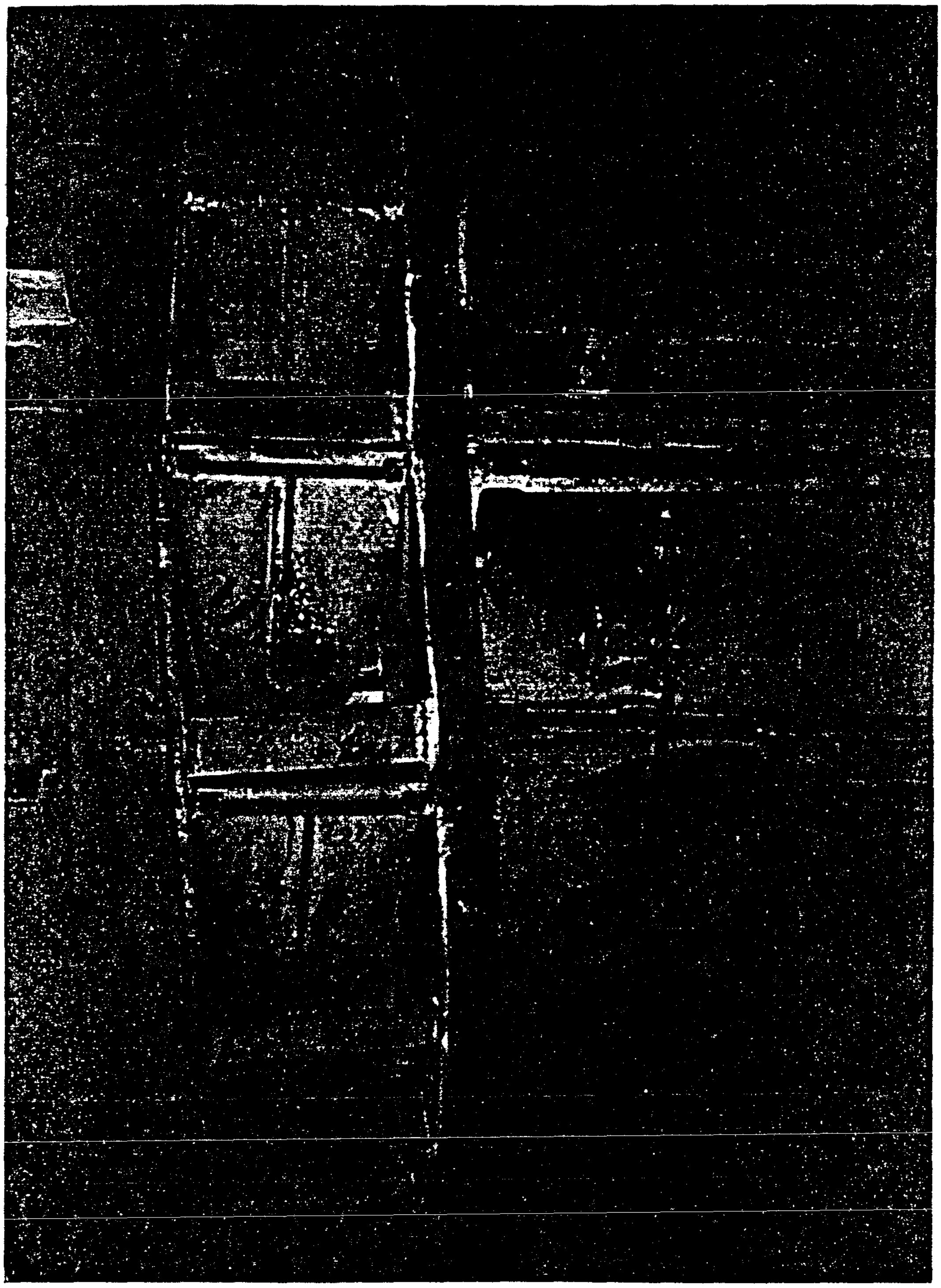


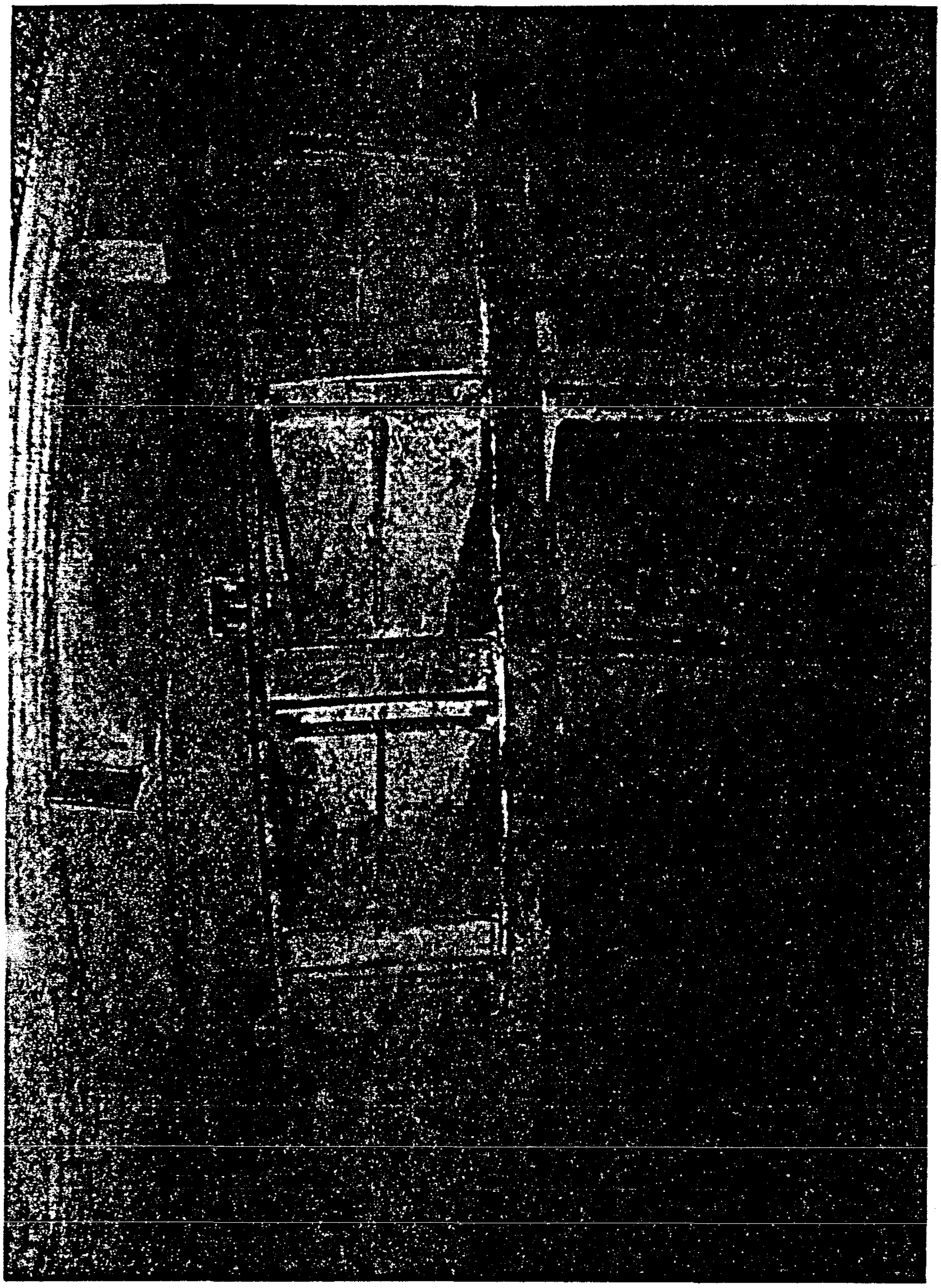


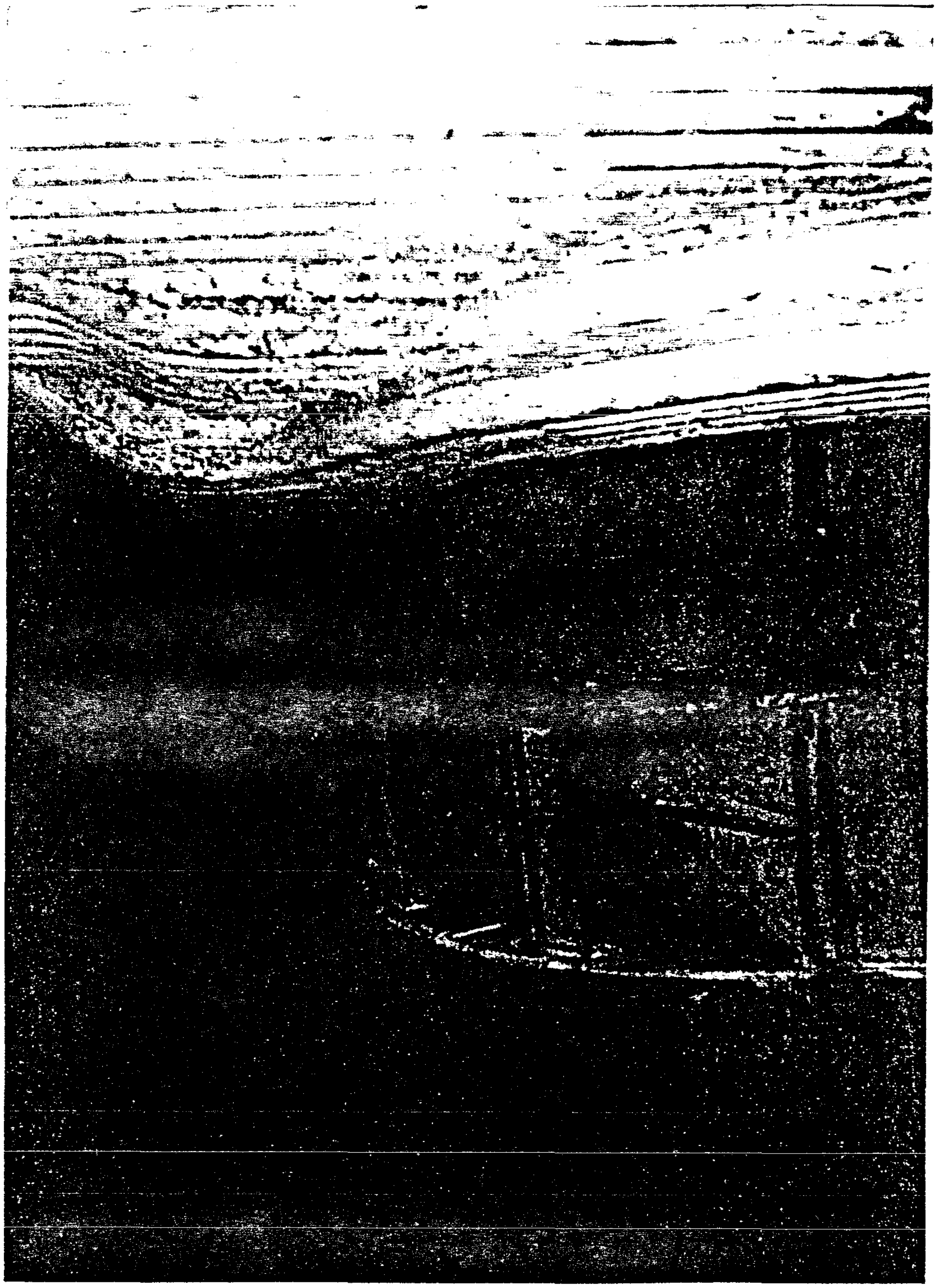


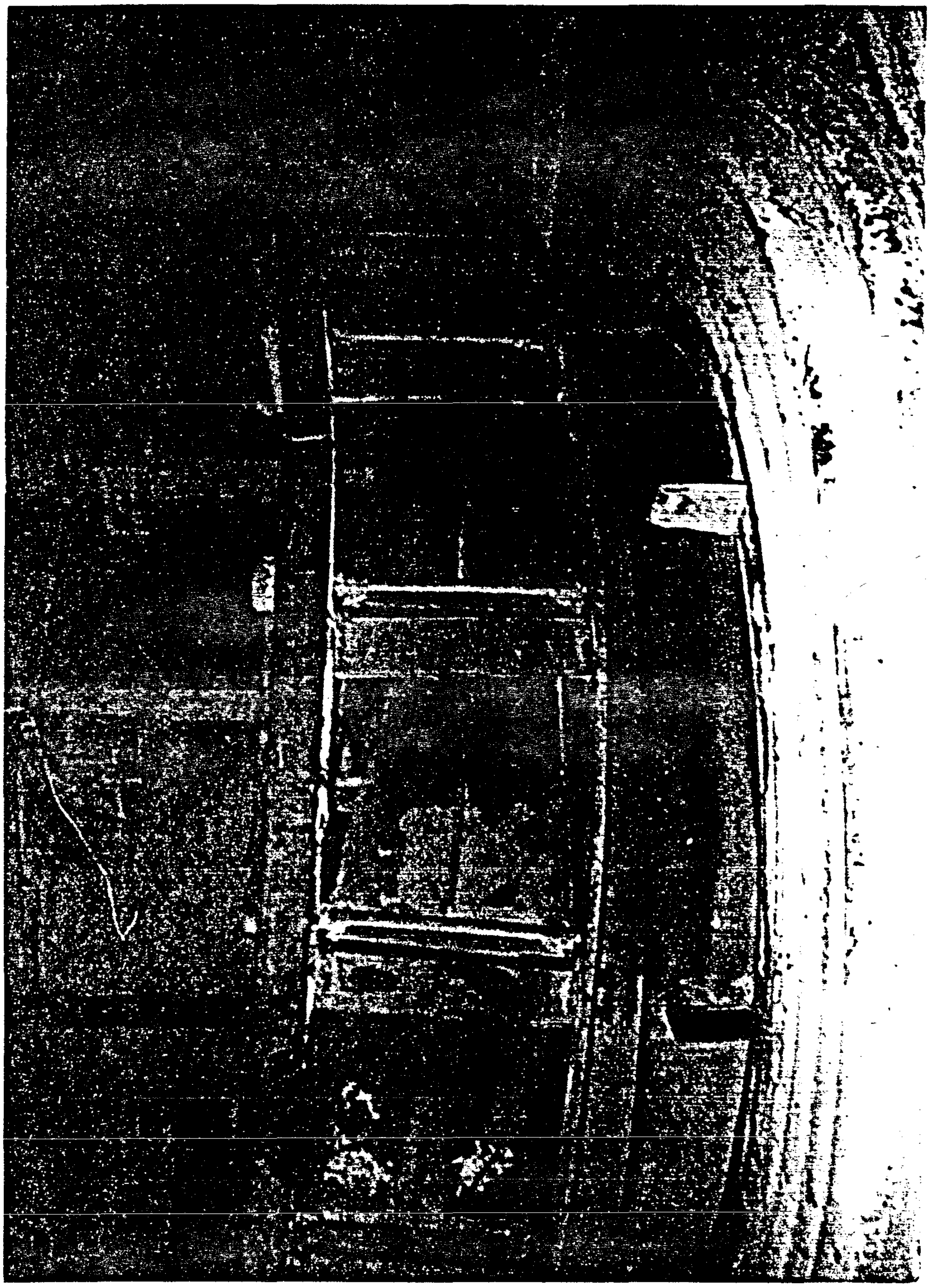


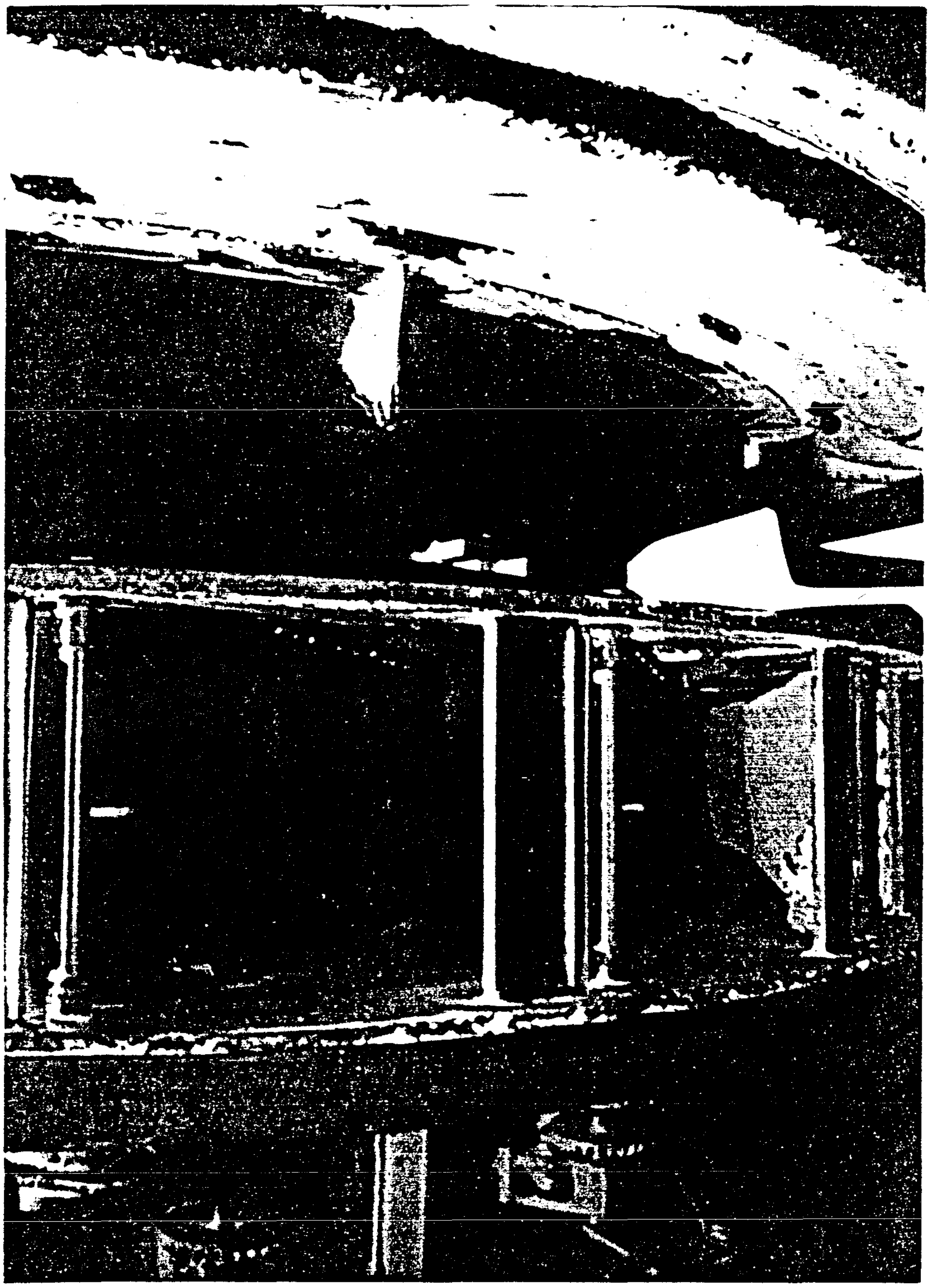












APPENDIX II

AERODYNAMIC CALCULATIONS

- **Baseline Existing Design**
  ΔP = 2.0 inch of water

- **Swirler Design Case (a) No Band**
  ΔP = 1.19 inch of water

- **Swirler Design Case (b) With Band**
  ΔP = 1.99 inch of water

RADIAL AND AXIAL BURNER AIR SWIRL SYSTEM
CONSTANT DENSITY, ADIABATIC, ISENTROPIC
FREE VORTEX IN ANNULAR SPACE, FORCED VORTEX IN SWIRL VANE
LINEAR SWIRL VANE EXIT ANGLE VERSUS RADIUS
PLUME OUTER (TIP) STREAMLINE EQUAL TO EXIT PRESSURE
VERSION 1.54 4/06/89

Baseline

IPP, OUTER REGISTER OPEN 25 DEG (EXISTING)

RADIAL FLOW DAMPER VANE EXIT GEOMETRY
NUMBER OF VANES= 16
THICKNESS,FT= .01042
SPAN LENGTH,FT= .83330
RADIUS,FT= 2.91670
ANGLE (+C.W.LOOKING UPSTREAM),DEG= 65.00000

AXIAL FLOW ANNULAR SPACE AND SWIRL VANE EXIT GEOMETRY
NUMBER OF VANES= 0
THICKNESS,FT= .00000
ANNULAR SPACE TIP RADIUS,FT= 2.41670
ANNULAR SPACE HUB AND VANE TIP (SPLITTER) RADIUS,FT= 1.68750
VANE HUB RADIUS,FT= 1.68740
VANE ANGLE (+C.W.LOOKING UPSTREAM) VS.SPAN (1.0 IS SPLITTER), (DEG/DECIMAL)
(.00/1.0000) (.00/ .0000) (

DELTA P EQUALS 2.0 INCH H2O

BURNER SWIRL SYSTEM CONDITIONS
INLET TEMPERATURE,DEG F= 650.00000
INLET PRESSURE,PSFA=2126.62000
EXIT PRESSURE,PSFA=2116.22000
VANE CASCADE DYNAMIC HEAD LOSS,"q"= 1.00000
ANNULAR SPACE DYNAMIC HEAD LOSS,"q"= 1.00000

RADIAL DAMPER VANE EXIT CONDITIONS

R= 2.91670 PT=2126.62000 PS=2122.67500 TT=1109.70000 TS=1109.11100
V= 84.06414 VR= 35.52711 VT= 76.18795 AT= 65.00000 THROAT= 6.31497

FLOW,LBM/SEC= 19.06265
CIRCULATION,SQ FT/SEC=1396.23200

AXIAL ANNULAR SPACE EXIT CONDITIONS

STREAMLINE R= 2.4167 PT=2122.7 PS=2116.2 TT=1109.7 TS=1108.7 RHO= .0360
V= 107.6 VZ= 56.0 VT= 92.0 A= 58.68 B= 58.68 P= .00 TAU= .0000

STREAMTUBE R= 2.3805 DM= 2.1948 DGT= .152E+02 DGX= .382E+01 DSWIRL= 1.643
DGP= -.160E+00 DRECIR= 3.4

STREAMLINE R= 2.3438 PT=2122.7 PS=2115.9 TT=1109.7 TS=1108.7 RHO= .0360
V= 110.1 VZ= 56.0 VT= 94.8 A= 59.44 B= 59.44 P= .00 TAU= .0000

STREAMTUBE R= 2.3076 DM= 2.1284 DGT= .147E+02 DGX= .370E+01 DSWIRL= 1.642
DGP= -.486E+00 DRECIR= 3.0

STREAMLINE R= 2.2709 PT=2122.7 PS=2115.6 TT=1109.7 TS=1108.6 RHO= .0360
V= 112.7 VZ= 56.0 VT= 97.9 A= 60.22 B= 60.22 P= .00 TAU= .0000

STREAMTUBE R= 2.2347 DM= 2.0619 DGT= .142E+02 DGX= .359E+01 DSWIRL= 1.642
DGP= -.823E+00 DRECIR= 2.7

STREAMLINE R= 2.1979 PT=2122.7 PS=2115.2 TT=1109.7 TS=1108.6 RHO= .0360
V= 115.6 VZ= 56.0 VT= 101.1 A= 61.00 B= 61.00 P= .00 TAU= .0000

STREAMTUBE R= 2.1618 DM= 1.9958 DGT= .138E+02 DGX= .348E+01 DSWIRL= 1.640
DGP= -.117E+01 DRECIR= 2.3

STREAMLINE R= 2.1250 PT=2122.7 PS=2114.8 TT=1109.7 TS=1108.5 RHO= .0360
V= 118.7 VZ= 56.1 VT= 104.6 A= 61.80 B= 61.80 P= .00 TAU= .0000

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STREAMTUBE R= 2.0889 DM= 1.9295 DGT=  .133E+02 DGX=  .336E+01 DSWIRL=  1.640
                                DGP= -.153E+01 DRECIR=    1.9

STREAMLINE R= 2.0521 PT=2122.7 PS=2114.4 TT=1109.7 TS=1108.5 RHO= .0360
  V= 122.0 VZ=  56.1 VT= 108.3  A= 62.62  B= 62.62  P=   .00 TAU= .0000

STREAMTUBE R= 2.0160 DM= 1.8631 DGT=  .129E+02 DGX=  .325E+01 DSWIRL=  1.639
                                DGP= -.191E+01 DRECIR=    1.4

STREAMLINE R= 1.9792 PT=2122.7 PS=2113.9 TT=1109.7 TS=1108.4 RHO= .0360
  V= 125.5 VZ=  56.1 VT= 112.3  A= 63.44  B= 63.44  P=   .00 TAU= .0000

STREAMTUBE R= 1.9431 DM= 1.7972 DGT=  .124E+02 DGX=  .314E+01 DSWIRL=  1.637
                                DGP= -.230E+01 DRECIR=     .9

STREAMLINE R= 1.9063 PT=2122.7 PS=2113.4 TT=1109.7 TS=1108.3 RHO= .0360
  V= 129.4 VZ=  56.2 VT= 116.6  A= 64.27  B= 64.27  P=   .00 TAU= .0000

STREAMTUBE R= 1.8702 DM= 1.7311 DGT=  .120E+02 DGX=  .302E+01 DSWIRL=  1.636
                                DGP= -.272E+01 DRECIR=     .4

STREAMLINE R= 1.8333 PT=2122.7 PS=2112.7 TT=1109.7 TS=1108.2 RHO= .0360
  V= 133.6 VZ=  56.2 VT= 121.2  A= 65.11  B= 65.11  P=   .00 TAU= .0000

STREAMTUBE R= 1.7972 DM= 1.6652 DGT=  .115E+02 DGX=  .291E+01 DSWIRL=  1.635
                                DGP= -.315E+01 DRECIR=    -.3

STREAMLINE R= 1.7604 PT=2122.7 PS=2112.0 TT=1109.7 TS=1108.1 RHO= .0360
  V= 138.2 VZ=  56.3 VT= 126.2  A= 65.97  B= 65.97  P=   .00 TAU= .0000

STREAMTUBE R= 1.7243 DM= 1.5993 DGT=  .110E+02 DGX=  .280E+01 DSWIRL=  1.633
                                DGP= -.360E+01 DRECIR=   -1.0

STREAMLINE R= 1.6875 PT=2122.7 PS=2111.3 TT=1109.7 TS=1108.0 RHO= .0360
  V= 143.2 VZ=  56.3 VT= 131.7  A= 66.83  B= 66.83  P=   .00 TAU= .0000

FLOW,LBM/SEC=  18.96632
SWIRL NUMBER= 1.639  REFERRED TO RADIUS,FT= 2.41670
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2116.2


AXIAL SWIRL VANE EXIT CONDITIONS


STREAMLINE R= 1.6875 PT=2122.7 PS=2111.3 TT=1109.7 TS=1108.0 RHO= .0360
  V= 143.2 VZ= 143.2 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

STREAMTUBE R= 1.6874 DM=  .0055 DGT=  .000E+00 DGX=  .243E-01 DSWIRL=   .000
                                DGP= -.526E-02 DRECIR=   18.0

STREAMLINE R= 1.6874 PT=2122.7 PS=2111.3 TT=1109.7 TS=1108.0 RHO= .0360
  V= 143.2 VZ= 143.2 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

FLOW,LBM/SEC=    .00546
SWIRL NUMBER=  .000  REFERRED TO RADIUS,FT= 1.68750
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2116.2


CONVERGED, NUMBER OF FLOW ITERATIONS=       21

SWIRLER TO TOTAL FLOW RATIO=      .000


SYMBOL DEFINITION FOR EXIT CONDITION TABLES
    R - RADIUS,FT
    PT,PS - TOTAL,STATIC PRESSURE,PSFA
    TT,TS - TOTAL,STATIC TEMPERATURE,DEG R
    V,VZ,VT,VR - TOTAL,AXIAL,TANGENTIAL,RADIAL VELOCITY,FT/SEC
    A - PROJECTED FLOW ANGLE (ON PLANE PERP TO RADIAL LINE),DEG
    B - FLOW ANGLE (MERIDIONAL PLANE),DEG
    P - STREAMLINE SLOPE,DEG
    THROAT -AREA,SQ FT
    AT - THROAT ANGLE (OFF RADIAL LINE IN PLANE PERP TO C-L),DEG
    RHO - DENSITY,LBM/CU FT
    TAU - BLOCKAGE,FRACTION


SYMBOL DEFINITION FOR INCREMENTAL VALUES
    DM - INCREMENTAL MASS FLOW,LBM/SEC
    DGT - INCREMENTAL TANGENTIAL MOMENTUM,FT LBF
```

DGX - INCREMENTAL AXIAL MOMENTUM,LBF
DSWIRL - SWIRL NUMBER FOR INCREMENT,DIMENSIONLESS
DGP - INCREMENTAL AXIAL PRESSURE FORCE,LBF
DRECIR - RECIRCULATION PARAMETER FOR INCREMENT,PSFA
- (NEGATIVE MEANS RECIRCULATION ZONE)

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RADIAL AND AXIAL BURNER AIR SWIRL SYSTEM
CONSTANT DENSITY, ADIABATIC, ISENTROPIC
FREE VORTEX IN ANNULAR SPACE, FORCED VORTEX IN SWIRL VANE
LINEAR SWIRL VANE EXIT ANGLE VERSUS RADIUS
PLUME OUTER (TIP) STREAMLINE EQUAL TO EXIT PRESSURE
VERSION 1.54  4/06/89


IPP, INNER SPIN OPEN 30 DEG, SLIDE 5 INCH OPEN (EXISTING)


RADIAL FLOW DAMPER VANE EXIT GEOMETRY
    NUMBER OF VANES=         0
    THICKNESS,FT=    .00000
    SPAN LENGTH,FT=    .41660
    RADIUS,FT=   1.66670
    ANGLE (+C.W.LOOKING UPSTREAM),DEG=    .00000

AXIAL FLOW ANNULAR SPACE AND SWIRL VANE EXIT GEOMETRY
    NUMBER OF VANES=         8
    THICKNESS,FT=    .01042
    ANNULAR SPACE TIP RADIUS,FT=   1.66670
    ANNULAR SPACE HUB AND VANE TIP (SPLITTER) RADIUS,FT=   1.66660
    VANE HUB RADIUS,FT=    .91660
    VANE ANGLE (+C.W.LOOKING UPSTREAM) VS.SPAN (1.0 IS SPLITTER), (DEG/DECIMAL)
    ( 60.00/1.0000)     ( 60.00/ .0000)     (


DELTA P EQUALS 2.0 INCH H2O (OUTER REGISTER OPEN 25 DEG)


BURNER SWIRL SYSTEM CONDITIONS
    INLET TEMPERATURE,DEG F= 650.00000
    INLET PRESSURE,PSFA=2126.62000
    EXIT PRESSURE,PSFA=2111.30000
    VANE CASCADE DYNAMIC HEAD LOSS,"q"=   1.00000
    ANNULAR SPACE DYNAMIC HEAD LOSS,"q"=   1.00000


RADIAL DAMPER VANE EXIT CONDITIONS


R=   1.66670  PT=2126.62000  PS=2121.14800  TT=1109.70000  TS=1108.88300
V=  99.00971  VR=  99.00971  VT=    .00000  AT=    .00000  THROAT=   4.36271

FLOW,LBM/SEC=  15.50266
CIRCULATION,SQ FT/SEC=    .00000


AXIAL ANNULAR SPACE EXIT CONDITIONS


STREAMLINE R= 1.6667 PT=2121.1 PS=2111.3 TT=1109.7 TS=1108.2 RHO= .0360
  V= 133.0 VZ= 133.0 VT=    .0  A=    .00  B=    .00  P=    .00 TAU= .0000

STREAMTUBE R= 1.6666 DM=  .0050 DGT=  .000E+00 DGX=  .207E-01 DSWIRL=   .000
                                DGP=  .000E+00 DRECIR=   19.8  (14.9)

STREAMLINE R= 1.6666 PT=2121.1 PS=2111.3 TT=1109.7 TS=1108.2 RHO= .0360
  V= 133.0 VZ= 133.0 VT=    .0  A=    .00  B=    .00  P=    .00 TAU= .0000

FLOW,LBM/SEC=    .00501
SWIRL NUMBER=  .000  REFERRED TO RADIUS,FT= 1.66670
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2111.3  (2116.22)


AXIAL SWIRL VANE EXIT CONDITIONS


STREAMLINE R= 1.6666 PT=2121.1 PS=2111.3 TT=1109.7 TS=1108.2 RHO= .0360
  V= 133.0 VZ=  66.5 VT= 115.2  A= 60.00  B= 60.00  P=    .00 TAU= .0159

STREAMTUBE R= 1.5934 DM= 3.5842 DGT=  .208E+02 DGX=  .753E+01 DSWIRL=  1.655
                                DGP= -.477E+00 DRECIR=    4.7  (-.22)

STREAMLINE R= 1.5166 PT=2121.1 PS=2110.6 TT=1109.7 TS=1108.1 RHO= .0360
  V= 137.5 VZ=  68.7 VT= 119.1  A= 60.00  B= 60.00  P=    .00 TAU= .0175

STREAMTUBE R= 1.4435 DM= 3.3537 DGT=  .182E+02 DGX=  .729E+01 DSWIRL=  1.500
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                                   DGP= -.141E+01 DRECIR=    4.3 (-.62)

STREAMLINE R= 1.3666 PT=2121.1 PS=2109.8 TT=1109.7 TS=1108.0 RHO= .0360
  V= 142.5 VZ=  71.3 VT= 123.4  A= 60.00  B= 60.00  P=   .00 TAU= .0194

STREAMTUBE R= 1.2938 DM= 3.1134 DGT=  .157E+02 DGX=  .703E+01 DSWIRL=  1.344
                                   DGP= -.230E+01 DRECIR=    3.9 (-1.02)

STREAMLINE R= 1.2166 PT=2121.1 PS=2108.9 TT=1109.7 TS=1107.9 RHO= .0360
  V= 148.3 VZ=  74.2 VT= 128.4  A= 60.00  B= 60.00  P=   .00 TAU= .0218

STREAMTUBE R= 1.1441 DM= 2.8617 DGT=  .133E+02 DGX=  .674E+01 DSWIRL=  1.188
                                   DGP= -.314E+01 DRECIR=    3.3 (-1.62)

STREAMLINE R= 1.0666 PT=2121.1 PS=2107.8 TT=1109.7 TS=1107.7 RHO= .0360
  V= 155.1 VZ=  77.5 VT= 134.3  A= 60.00  B= 60.00  P=   .00 TAU= .0249

STREAMTUBE R=  .9944 DM= 2.5962 DGT=  .110E+02 DGX=  .641E+01 DSWIRL=  1.033
                                   DGP= -.390E+01 DRECIR=    2.7 (-2.22)

STREAMLINE R=  .9166 PT=2121.1 PS=2106.4 TT=1109.7 TS=1107.5 RHO= .0360
  V= 163.1 VZ=  81.6 VT= 141.3  A= 60.00  B= 60.00  P=   .00 TAU= .0289

FLOW,LBM/SEC=  15.50926
SWIRL NUMBER= 1.356  REFERRED TO RADIUS,FT= 1.66660
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2111.3 (2116.22)


CONVERGED, NUMBER OF FLOW ITERATIONS=         6

SWIRLER TO TOTAL FLOW RATIO=     1.000


SYMBOL DEFINITION FOR EXIT CONDITION TABLES
    R - RADIUS,FT
    PT,PS - TOTAL,STATIC PRESSURE,PSFA
    TT,TS - TOTAL,STATIC TEMPERATURE,DEG R
    V,VZ,VT,VR - TOTAL,AXIAL,TANGENTIAL,RADIAL VELOCITY,FT/SEC
    A - PROJECTED FLOW ANGLE (ON PLANE PERP TO RADIAL LINE),DEG
    B - FLOW ANGLE (MERIDIONAL PLANE),DEG
    P - STREAMLINE SLOPE,DEG
    THROAT -AREA,SQ FT
    AT - THROAT ANGLE (OFF RADIAL LINE IN PLANE PERP TO C-L),DEG
    RHO - DENSITY,LBM/CU FT
    TAU - BLOCKAGE,FRACTION


SYMBOL DEFINITION FOR INCREMENTAL VALUES
    DM - INCREMENTAL MASS FLOW,LBM/SEC
    DGT - INCREMENTAL TANGENTIAL MOMENTUM,FT LBF
    DGX - INCREMENTAL AXIAL MOMENTUM,LBF
    DSWIRL - SWIRL NUMBER FOR INCREMENT,DIMENSIONLESS
    DGP - INCREMENTAL AXIAL PRESSURE FORCE,LBF
    DRECIR - RECIRCULATION PARAMETER FOR INCREMENT,PSFA
           - (NEGATIVE MEANS RECIRCULATION ZONE)
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RADIAL AND AXIAL BURNER AIR SWIRL SYSTEM                         Case A
CONSTANT DENSITY, ADIABATIC, ISENTROPIC
FREE VORTEX IN ANNULAR SPACE, FORCED VORTEX IN SWIRL VANE
LINEAR SWIRL VANE EXIT ANGLE VERSUS RADIUS
PLUME OUTER (TIP) STREAMLINE EQUAL TO EXIT PRESSURE
VERSION 1.54  4/06/89


IPP, OUTER REGISTER OPEN 34 DEG (EXISTING)


RADIAL FLOW DAMPER VANE EXIT GEOMETRY
    NUMBER OF VANES=        16
    THICKNESS,FT=    .01042
    SPAN LENGTH,FT=    .83330
    RADIUS,FT=   2.91670
    ANGLE (+C.W.LOOKING UPSTREAM),DEG=  56.00000

AXIAL FLOW ANNULAR SPACE AND SWIRL VANE EXIT GEOMETRY
    NUMBER OF VANES=         0
    THICKNESS,FT=    .00000
    ANNULAR SPACE TIP RADIUS,FT=   2.41670
    ANNULAR SPACE HUB AND VANE TIP (SPLITTER) RADIUS,FT=   1.68750
    VANE HUB RADIUS,FT=   1.68740
    VANE ANGLE (+C.W.LOOKING UPSTREAM) VS.SPAN (1.0 IS SPLITTER), (DEG/DECIMAL)
    (   .00/1.0000)     (   .00/ .0000)     (


DELTA P EQUALS 1.19 INCH H2O


BURNER SWIRL SYSTEM CONDITIONS
    INLET TEMPERATURE,DEG F= 650.00000
    INLET PRESSURE,PSFA=2122.42000
    EXIT PRESSURE,PSFA=2116.22000
    VANE CASCADE DYNAMIC HEAD LOSS,"q"=    1.00000
    ANNULAR SPACE DYNAMIC HEAD LOSS,"q"=    1.00000


RADIAL DAMPER VANE EXIT CONDITIONS


R=   2.91670  PT=2122.42000  PS=2120.19200  TT=1109.70000  TS=1109.36700
V=  63.22945  VR=  35.35750  VT=  52.41956  AT=  56.00000  THROAT=   8.40062

FLOW,LBM/SEC=  19.04467
CIRCULATION,SQ FT/SEC= 960.64870


AXIAL ANNULAR SPACE EXIT CONDITIONS


STREAMLINE R= 2.4167 PT=2120.2 PS=2116.2 TT=1109.7 TS=1109.1 RHO= .0359
  V=  84.5 VZ=  56.0 VT=  63.3  A= 48.50  B= 48.50  P=   .00 TAU= .0000

STREAMTUBE R= 2.3805 DM= 2.1906 DGT=  .104E+02 DGX=  .381E+01 DSWIRL=  1.130
                                DGP= -.758E-01 DRECIR=    3.4

STREAMLINE R= 2.3438 PT=2120.2 PS=2116.1 TT=1109.7 TS=1109.1 RHO= .0359
  V=  86.0 VZ=  56.0 VT=  65.2  A= 49.37  B= 49.37  P=   .00 TAU= .0000

STREAMTUBE R= 2.3076 DM= 2.1237 DGT=  .101E+02 DGX=  .369E+01 DSWIRL=  1.130
                                DGP= -.230E+00 DRECIR=    3.3

STREAMLINE R= 2.2709 PT=2120.2 PS=2115.9 TT=1109.7 TS=1109.1 RHO= .0359
  V=  87.6 VZ=  56.0 VT=  67.3  A= 50.25  B= 50.25  P=   .00 TAU= .0000

STREAMTUBE R= 2.2347 DM= 2.0568 DGT=  .977E+01 DGX=  .358E+01 DSWIRL=  1.130
                                DGP= -.389E+00 DRECIR=    3.1

STREAMLINE R= 2.1979 PT=2120.2 PS=2115.8 TT=1109.7 TS=1109.0 RHO= .0359
  V=  89.3 VZ=  56.0 VT=  69.6  A= 51.17  B= 51.17  P=   .00 TAU= .0000

STREAMTUBE R= 2.1618 DM= 1.9899 DGT=  .945E+01 DGX=  .346E+01 DSWIRL=  1.130
                                DGP= -.553E+00 DRECIR=    2.9

STREAMLINE R= 2.1250 PT=2120.2 PS=2115.6 TT=1109.7 TS=1109.0 RHO= .0359
  V=  91.2 VZ=  56.0 VT=  71.9  A= 52.11  B= 52.11  P=   .00 TAU= .0000
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STREAMTUBE R= 2.0889 DM= 1.9230 DGT=  .914E+01 DGX=  .335E+01 DSWIRL=  1.130
                                DGP= -.724E+00 DRECIR=    2.7

STREAMLINE R= 2.0521 PT=2120.2 PS=2115.4 TT=1109.7 TS=1109.0 RHO= .0359
  V=  93.2 VZ=  56.0 VT=  74.5  A= 53.07  B= 53.07  P=   .00 TAU= .0000

STREAMTUBE R= 2.0160 DM= 1.8561 DGT=  .882E+01 DGX=  .323E+01 DSWIRL=  1.130
                                DGP= -.903E+00 DRECIR=    2.5

STREAMLINE R= 1.9792 PT=2120.2 PS=2115.1 TT=1109.7 TS=1108.9 RHO= .0359
  V=  95.4 VZ=  56.0 VT=  77.3  A= 54.05  B= 54.05  P=   .00 TAU= .0000

STREAMTUBE R= 1.9431 DM= 1.7892 DGT=  .850E+01 DGX=  .311E+01 DSWIRL=  1.129
                                DGP= -.109E+01 DRECIR=    2.3

STREAMLINE R= 1.9063 PT=2120.2 PS=2114.9 TT=1109.7 TS=1108.9 RHO= .0359
  V=  97.8 VZ=  56.0 VT=  80.2  A= 55.07  B= 55.07  P=   .00 TAU= .0000

STREAMTUBE R= 1.8702 DM= 1.7224 DGT=  .818E+01 DGX=  .300E+01 DSWIRL=  1.129
                                DGP= -.128E+01 DRECIR=    2.0

STREAMLINE R= 1.8333 PT=2120.2 PS=2114.6 TT=1109.7 TS=1108.9 RHO= .0359
  V= 100.5 VZ=  56.0 VT=  83.4  A= 56.10  B= 56.10  P=   .00 TAU= .0000

STREAMTUBE R= 1.7972 DM= 1.6557 DGT=  .787E+01 DGX=  .288E+01 DSWIRL=  1.129
                                DGP= -.149E+01 DRECIR=    1.7

STREAMLINE R= 1.7604 PT=2120.2 PS=2114.2 TT=1109.7 TS=1108.8 RHO= .0359
  V= 103.4 VZ=  56.1 VT=  86.8  A= 57.16  B= 57.16  P=   .00 TAU= .0000

STREAMTUBE R= 1.7243 DM= 1.5891 DGT=  .755E+01 DGX=  .277E+01 DSWIRL=  1.128
                                DGP= -.170E+01 DRECIR=    1.3

STREAMLINE R= 1.6875 PT=2120.2 PS=2113.9 TT=1109.7 TS=1108.8 RHO= .0359
  V= 106.6 VZ=  56.1 VT=  90.6  A= 58.25  B= 58.25  P=   .00 TAU= .0000

FLOW,LBM/SEC=  18.89652
SWIRL NUMBER= 1.130  REFERRED TO RADIUS,FT= 2.41670
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2116.2


AXIAL SWIRL VANE EXIT CONDITIONS


STREAMLINE R= 1.6875 PT=2120.2 PS=2113.9 TT=1109.7 TS=1108.8 RHO= .0359
  V= 106.6 VZ= 106.6 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

STREAMTUBE R= 1.6874 DM=  .0041 DGT=  .000E+00 DGX=  .134E-01 DSWIRL=   .000
                                DGP= -.249E-02 DRECIR=   10.3

STREAMLINE R= 1.6874 PT=2120.2 PS=2113.9 TT=1109.7 TS=1108.8 RHO= .0359
  V= 106.6 VZ= 106.6 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

FLOW,LBM/SEC=     .00405
SWIRL NUMBER=  .000  REFERRED TO RADIUS,FT= 1.68750
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2116.2


CONVERGED, NUMBER OF FLOW ITERATIONS=        29

SWIRLER TO TOTAL FLOW RATIO=      .000


SYMBOL DEFINITION FOR EXIT CONDITION TABLES
    R - RADIUS,FT
    PT,PS - TOTAL,STATIC PRESSURE,PSFA
    TT,TS - TOTAL,STATIC TEMPERATURE,DEG R
    V,VZ,VT,VR - TOTAL,AXIAL,TANGENTIAL,RADIAL VELOCITY,FT/SEC
    A - PROJECTED FLOW ANGLE (ON PLANE PERP TO RADIAL LINE),DEG
    B - FLOW ANGLE (MERIDIONAL PLANE),DEG
    P - STREAMLINE SLOPE,DEG
    THROAT -AREA,SQ FT
    AT - THROAT ANGLE (OFF RADIAL LINE IN PLANE PERP TO C-L),DEG
    RHO - DENSITY,LBM/CU FT
    TAU - BLOCKAGE,FRACTION


SYMBOL DEFINITION FOR INCREMENTAL VALUES
    DM - INCREMENTAL MASS FLOW,LBM/SEC
    DGT - INCREMENTAL TANGENTIAL MOMENTUM,FT LBF
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DGX - INCREMENTAL AXIAL MOMENTUM,LBF
DSWIRL - SWIRL NUMBER FOR INCREMENT,DIMENSIONLESS
DGP - INCREMENTAL AXIAL PRESSURE FORCE,LBF
DRECIR - RECIRCULATION PARAMETER FOR INCREMENT,PSFA
- (NEGATIVE MEANS RECIRCULATION ZONE)

Case A

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RADIAL AND AXIAL BURNER AIR SWIRL SYSTEM
CONSTANT DENSITY, ADIABATIC, ISENTROPIC
FREE VORTEX IN ANNULAR SPACE, FORCED VORTEX IN SWIRL VANE
LINEAR SWIRL VANE EXIT ANGLE VERSUS RADIUS
PLUME OUTER (TIP) STREAMLINE EQUAL TO EXIT PRESSURE
VERSION 1.54  4/06/89


IPP, INNER SPIN OPEN 90 DEG, SLIDE 10 INCH OPEN, 40 INCH SWIRLER


RADIAL FLOW DAMPER VANE EXIT GEOMETRY
    NUMBER OF VANES=         0
    THICKNESS,FT=    .00000
    SPAN LENGTH,FT=    .83330
    RADIUS,FT=   1.66670
    ANGLE (+C.W.LOOKING UPSTREAM),DEG=    .00000

AXIAL FLOW ANNULAR SPACE AND SWIRL VANE EXIT GEOMETRY
    NUMBER OF VANES=        40
    THICKNESS,FT=    .01042
    ANNULAR SPACE TIP RADIUS,FT=   1.66670
    ANNULAR SPACE HUB AND VANE TIP (SPLITTER) RADIUS,FT=   1.66660
    VANE HUB RADIUS,FT=    .91660
    VANE ANGLE (+C.W.LOOKING UPSTREAM) VS.SPAN (1.0 IS SPLITTER), (DEG/DECIMAL)
    ( 64.00/1.0000)     ( 40.00/ .1111)     (    .00/ .0000)     (


DELTA P EQUALS 1.19 INCH H2O (OUTER REGISTER OPEN 34 DEG)


BURNER SWIRL SYSTEM CONDITIONS
    INLET TEMPERATURE,DEG F= 650.00000
    INLET PRESSURE,PSFA=2122.42000
    EXIT PRESSURE,PSFA=2113.90000
    VANE CASCADE DYNAMIC HEAD LOSS,"q"=   1.00000
    ANNULAR SPACE DYNAMIC HEAD LOSS,"q"=   1.00000


RADIAL DAMPER VANE EXIT CONDITIONS


R=   1.66670  PT=2122.42000  PS=2121.05900  TT=1109.70000  TS=1109.49700
V=  49.39742  VR=  49.39742  VT=    .00000  AT=    .00000  THROAT=   8.72646

FLOW,LBM/SEC=  15.46160
CIRCULATION,SQ FT/SEC=    .00000


AXIAL ANNULAR SPACE EXIT CONDITIONS


STREAMLINE R= 1.6667 PT=2121.1 PS=2113.9 TT=1109.7 TS=1108.6 RHO= .0359
  V= 113.4 VZ= 113.4 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

STREAMTUBE R= 1.6666 DM=  .0043 DGT=  .000E+00 DGX=  .150E-01 DSWIRL=   .000
                              DGP=  .000E+00 DRECIR=   14.3  (12.0)

STREAMLINE R= 1.6666 PT=2121.1 PS=2113.9 TT=1109.7 TS=1108.6 RHO= .0359
  V= 113.4 VZ= 113.4 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

FLOW,LBM/SEC=    .00426
SWIRL NUMBER=  .000  REFERRED TO RADIUS,FT= 1.66670
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2113.9 (2116.22)


AXIAL SWIRL VANE EXIT CONDITIONS


STREAMLINE R= 1.6666 PT=2121.1 PS=2113.9 TT=1109.7 TS=1108.6 RHO= .0359
  V= 113.4 VZ=  49.7 VT= 101.9  A= 64.00  B= 64.00  P=   .00 TAU= .0908

STREAMTUBE R= 1.6255 DM= 1.4700 DGT=  .753E+01 DGX=  .242E+01 DSWIRL=  1.866
                              DGP= -.120E+00 DRECIR=    2.7  (.38)

STREAMLINE R= 1.5833 PT=2121.1 PS=2113.6 TT=1109.7 TS=1108.6 RHO= .0359
  V= 115.7 VZ=  56.1 VT= 101.2  A= 61.00  B= 61.00  P=   .00 TAU= .0864

STREAMTUBE R= 1.5422 DM= 1.5699 DGT=  .756E+01 DGX=  .290E+01 DSWIRL=  1.566
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                                   DGP= -.352E+00 DRECIR=    3.2  (.88)

STREAMLINE R= 1.4999 PT=2121.1 PS=2113.3 TT=1109.7 TS=1108.5 RHO= .0359
  V= 118.0 VZ=  62.5 VT= 100.1  A= 58.00  B= 58.00  P=   .00 TAU= .0835

STREAMTUBE R= 1.4589 DM= 1.6513 DGT=  .743E+01 DGX=  .338E+01 DSWIRL=  1.319
                                   DGP= -.566E+00 DRECIR=    3.7  (1.38)

STREAMLINE R= 1.4166 PT=2121.1 PS=2113.0 TT=1109.7 TS=1108.5 RHO= .0359
  V= 120.3 VZ=  69.0 VT=  98.6  A= 55.00  B= 55.00  P=   .00 TAU= .0816

STREAMTUBE R= 1.3756 DM= 1.7134 DGT=  .715E+01 DGX=  .385E+01 DSWIRL=  1.113
                                   DGP= -.760E+00 DRECIR=    4.3  (1.98)

STREAMLINE R= 1.3333 PT=2121.1 PS=2112.7 TT=1109.7 TS=1108.4 RHO= .0359
  V= 122.7 VZ=  75.5 VT=  96.7  A= 52.00  B= 52.00  P=   .00 TAU= .0808

STREAMTUBE R= 1.2923 DM= 1.7554 DGT=  .673E+01 DGX=  .430E+01 DSWIRL=   .939
                                   DGP= -.931E+00 DRECIR=    5.0  (2.68)

STREAMLINE R= 1.2499 PT=2121.1 PS=2112.4 TT=1109.7 TS=1108.4 RHO= .0359
  V= 125.1 VZ=  82.1 VT=  94.4  A= 49.00  B= 49.00  P=   .00 TAU= .0809

STREAMTUBE R= 1.2090 DM= 1.7764 DGT=  .621E+01 DGX=  .471E+01 DSWIRL=   .791
                                   DGP= -.108E+01 DRECIR=    5.7  (3.38)

STREAMLINE R= 1.1666 PT=2121.1 PS=2112.0 TT=1109.7 TS=1108.3 RHO= .0359
  V= 127.4 VZ=  88.5 VT=  91.7  A= 46.00  B= 46.00  P=   .00 TAU= .0819

STREAMTUBE R= 1.1257 DM= 1.7758 DGT=  .560E+01 DGX=  .506E+01 DSWIRL=   .664
                                   DGP= -.120E+01 DRECIR=    6.6  (4.28)

STREAMLINE R= 1.0833 PT=2121.1 PS=2111.7 TT=1109.7 TS=1108.3 RHO= .0359
  V= 129.8 VZ=  94.9 VT=  88.5  A= 43.00  B= 43.00  P=   .00 TAU= .0837

STREAMTUBE R= 1.0424 DM= 1.7533 DGT=  .493E+01 DGX=  .534E+01 DSWIRL=   .554
                                   DGP= -.129E+01 DRECIR=    7.4  (5.08)

STREAMLINE R=  .9999 PT=2121.1 PS=2111.4 TT=1109.7 TS=1108.2 RHO= .0359
  V= 132.1 VZ= 101.2 VT=  84.9  A= 40.00  B= 40.00  P=   .00 TAU= .0866

STREAMTUBE R=  .9592 DM= 1.9414 DGT=  .229E+01 DGX=  .721E+01 DSWIRL=   .191
                                   DGP= -.136E+01 DRECIR=   11.7  (9.38)

STREAMLINE R=  .9166 PT=2121.1 PS=2111.0 TT=1109.7 TS=1108.2 RHO= .0359
  V= 134.4 VZ= 134.4 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0724

FLOW,LBM/SEC=  15.40694
SWIRL NUMBER=  .849  REFERRED TO RADIUS,FT= 1.66660
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2113.9 (2116.22)

CONVERGED, NUMBER OF FLOW ITERATIONS=        7

SWIRLER TO TOTAL FLOW RATIO=     1.000


SYMBOL DEFINITION FOR EXIT CONDITION TABLES
    R - RADIUS,FT
    PT,PS - TOTAL,STATIC PRESSURE,PSFA
    TT,TS - TOTAL,STATIC TEMPERATURE,DEG R
    V,VZ,VT,VR - TOTAL,AXIAL,TANGENTIAL,RADIAL VELOCITY,FT/SEC
    A - PROJECTED FLOW ANGLE (ON PLANE PERP TO RADIAL LINE),DEG
    B - FLOW ANGLE (MERIDIONAL PLANE),DEG
    P - STREAMLINE SLOPE,DEG
    THROAT -AREA,SQ FT
    AT - THROAT ANGLE (OFF RADIAL LINE IN PLANE PERP TO C-L),DEG
    RHO - DENSITY,LBM/CU FT
    TAU - BLOCKAGE,FRACTION


SYMBOL DEFINITION FOR INCREMENTAL VALUES
    DM - INCREMENTAL MASS FLOW,LBM/SEC
    DGT - INCREMENTAL TANGENTIAL MOMENTUM,FT LBF
    DGX - INCREMENTAL AXIAL MOMENTUM,LBF
    DSWIRL - SWIRL NUMBER FOR INCREMENT,DIMENSIONLESS
    DGP - INCREMENTAL AXIAL PRESSURE FORCE,LBF
    DRECIR - RECIRCULATION PARAMETER FOR INCREMENT,PSFA
           - (NEGATIVE MEANS RECIRCULATION ZONE)
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RADIAL AND AXIAL BURNER AIR SWIRL SYSTEM
CONSTANT DENSITY, ADIABATIC, ISENTROPIC
FREE VORTEX IN ANNULAR SPACE, FORCED VORTEX IN SWIRL VANE
LINEAR SWIRL VANE EXIT ANGLE VERSUS RADIUS
PLUME OUTER (TIP) STREAMLINE EQUAL TO EXIT PRESSURE
VERSION 1.54  4/06/89


IPP, OUTER REGISTER OPEN 40 DEG (EXISTING) W/BAND 1304 SQ IN OPEN INLET


RADIAL FLOW DAMPER VANE EXIT GEOMETRY
    NUMBER OF VANES=        16
    THICKNESS,FT=     .01042
    SPAN LENGTH,FT=     .83330
    RADIUS,FT=   2.91670
    ANGLE (+C.W.LOOKING UPSTREAM),DEG=  50.00000

AXIAL FLOW ANNULAR SPACE AND SWIRL VANE EXIT GEOMETRY
    NUMBER OF VANES=        0
    THICKNESS,FT=     .00000
    ANNULAR SPACE TIP RADIUS,FT=   2.41670
    ANNULAR SPACE HUB AND VANE TIP (SPLITTER) RADIUS,FT=   1.68750
    VANE HUB RADIUS,FT=   1.68740
    VANE ANGLE (+C.W.LOOKING UPSTREAM) VS.SPAN (1.0 IS SPLITTER), (DEG/DECIMAL)
    (   .00/1.0000)     (   .00/ .0000)     (


DELTA P:TOTAL=1.99 INCH H2O; BAND=1.05 AND REG TO FURN=0.94


BURNER SWIRL SYSTEM CONDITIONS
    INLET TEMPERATURE,DEG F= 650.00000
    INLET PRESSURE,PSFA=2121.11000
    EXIT PRESSURE,PSFA=2116.22000
    VANE CASCADE DYNAMIC HEAD LOSS,"q"=   1.00000
    ANNULAR SPACE DYNAMIC HEAD LOSS,"q"=   1.00000


RADIAL DAMPER VANE EXIT CONDITIONS


R=   2.91670  PT=2121.11000  PS=2119.41900  TT=1109.70000  TS=1109.44700
V=  55.10514  VR=  35.42093  VT=  42.21296  AT=  50.00000  THROAT=   9.67721

FLOW,LBM/SEC=  19.10859
CIRCULATION,SQ FT/SEC= 773.60100


AXIAL ANNULAR SPACE EXIT CONDITIONS


STREAMLINE R= 2.4167 PT=2119.4 PS=2116.2 TT=1109.7 TS=1109.2 RHO= .0359
  V=  75.8 VZ=  56.1 VT=  50.9  A= 42.22  B= 42.22  P=   .00 TAU= .0000

STREAMTUBE R= 2.3805 DM= 2.1960 DGT=  .840E+01 DGX=  .383E+01 DSWIRL=   .907
                                DGP= -.490E-01 DRECIR=    3.5

STREAMLINE R= 2.3438 PT=2119.4 PS=2116.1 TT=1109.7 TS=1109.2 RHO= .0359
  V=  76.9 VZ=  56.2 VT=  52.5  A= 43.09  B= 43.09  P=   .00 TAU= .0000

STREAMTUBE R= 2.3076 DM= 2.1288 DGT=  .815E+01 DGX=  .372E+01 DSWIRL=   .907
                                DGP= -.149E+00 DRECIR=    3.4

STREAMLINE R= 2.2709 PT=2119.4 PS=2116.0 TT=1109.7 TS=1109.2 RHO= .0359
  V=  78.1 VZ=  56.1 VT=  54.2  A= 44.00  B= 44.00  P=   .00 TAU= .0000

STREAMTUBE R= 2.2347 DM= 2.0617 DGT=  .789E+01 DGX=  .360E+01 DSWIRL=   .907
                                DGP= -.252E+00 DRECIR=    3.3

STREAMLINE R= 2.1979 PT=2119.4 PS=2115.9 TT=1109.7 TS=1109.2 RHO= .0359
  V=  79.3 VZ=  56.2 VT=  56.0  A= 44.92  B= 44.92  P=   .00 TAU= .0000

STREAMTUBE R= 2.1618 DM= 1.9947 DGT=  .763E+01 DGX=  .348E+01 DSWIRL=   .907
                                DGP= -.359E+00 DRECIR=    3.2

STREAMLINE R= 2.1250 PT=2119.4 PS=2115.8 TT=1109.7 TS=1109.2 RHO= .0359
  V=  80.7 VZ=  56.2 VT=  57.9  A= 45.89  B= 45.89  P=   .00 TAU= .0000
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STREAMTUBE R= 2.0889 DM= 1.9274 DGT=  .737E+01 DGX=  .336E+01 DSWIRL=   .907
                                DGP= -.470E+00 DRECIR=    3.0

STREAMLINE R= 2.0521 PT=2119.4 PS=2115.7 TT=1109.7 TS=1109.1 RHO= .0359
  V=  82.2 VZ=  56.2 VT=  60.0  A= 46.89  B= 46.89  P=   .00 TAU= .0000

STREAMTUBE R= 2.0160 DM= 1.8603 DGT=  .712E+01 DGX=  .325E+01 DSWIRL=   .907
                                DGP= -.585E+00 DRECIR=    2.9

STREAMLINE R= 1.9792 PT=2119.4 PS=2115.5 TT=1109.7 TS=1109.1 RHO= .0359
  V=  83.8 VZ=  56.2 VT=  62.2  A= 47.92  B= 47.92  P=   .00 TAU= .0000

STREAMTUBE R= 1.9431 DM= 1.7934 DGT=  .686E+01 DGX=  .313E+01 DSWIRL=   .907
                                DGP= -.706E+00 DRECIR=    2.7

STREAMLINE R= 1.9063 PT=2119.4 PS=2115.3 TT=1109.7 TS=1109.1 RHO= .0359
  V=  85.6 VZ=  56.2 VT=  64.6  A= 48.98  B= 48.98  P=   .00 TAU= .0000

STREAMTUBE R= 1.8702 DM= 1.7261 DGT=  .660E+01 DGX=  .301E+01 DSWIRL=   .907
                                DGP= -.832E+00 DRECIR=    2.5

STREAMLINE R= 1.8333 PT=2119.4 PS=2115.2 TT=1109.7 TS=1109.1 RHO= .0359
  V=  87.6 VZ=  56.2 VT=  67.2  A= 50.09  B= 50.09  P=   .00 TAU= .0000

STREAMTUBE R= 1.7972 DM= 1.6588 DGT=  .635E+01 DGX=  .290E+01 DSWIRL=   .907
                                DGP= -.964E+00 DRECIR=    2.3

STREAMLINE R= 1.7604 PT=2119.4 PS=2114.9 TT=1109.7 TS=1109.0 RHO= .0359
  V=  89.7 VZ=  56.2 VT=  69.9  A= 51.22  B= 51.22  P=   .00 TAU= .0000

STREAMTUBE R= 1.7243 DM= 1.5918 DGT=  .609E+01 DGX=  .278E+01 DSWIRL=   .907
                                DGP= -.110E+01 DRECIR=    2.1

STREAMLINE R= 1.6875 PT=2119.4 PS=2114.7 TT=1109.7 TS=1109.0 RHO= .0359
  V=  92.1 VZ=  56.2 VT=  73.0  A= 52.39  B= 52.39  P=   .00 TAU= .0000

FLOW,LBM/SEC=  18.93924
SWIRL NUMBER=  .907  REFERRED TO RADIUS,FT= 2.41670
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2116.2


AXIAL SWIRL VANE EXIT CONDITIONS


STREAMLINE R= 1.6875 PT=2119.4 PS=2114.7 TT=1109.7 TS=1109.0 RHO= .0359
  V=  92.1 VZ=  92.1 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

STREAMTUBE R= 1.6874 DM=  .0035 DGT=  .000E+00 DGX=  .100E-01 DSWIRL=   .000
                                DGP= -.161E-02 DRECIR=    7.9

STREAMLINE R= 1.6874 PT=2119.4 PS=2114.7 TT=1109.7 TS=1109.0 RHO= .0359
  V=  92.1 VZ=  92.1 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

FLOW,LBM/SEC=     .00350
SWIRL NUMBER=  .000  REFERRED TO RADIUS,FT= 1.68750
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2116.2


CONVERGED, NUMBER OF FLOW ITERATIONS=        34

SWIRLER TO TOTAL FLOW RATIO=      .000


SYMBOL DEFINITION FOR EXIT CONDITION TABLES
    R - RADIUS,FT
    PT,PS - TOTAL,STATIC PRESSURE,PSFA
    TT,TS - TOTAL,STATIC TEMPERATURE,DEG R
    V,VZ,VT,VR - TOTAL,AXIAL,TANGENTIAL,RADIAL VELOCITY,FT/SEC
    A - PROJECTED FLOW ANGLE (ON PLANE PERP TO RADIAL LINE),DEG
    B - FLOW ANGLE (MERIDIONAL PLANE),DEG
    P - STREAMLINE SLOPE,DEG
    THROAT -AREA,SQ FT
    AT - THROAT ANGLE (OFF RADIAL LINE IN PLANE PERP TO C-L),DEG
    RHO - DENSITY,LBM/CU FT
    TAU - BLOCKAGE,FRACTION


SYMBOL DEFINITION FOR INCREMENTAL VALUES
    DM - INCREMENTAL MASS FLOW,LBM/SEC
    DGT - INCREMENTAL TANGENTIAL MOMENTUM,FT LBF
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DGX - INCREMENTAL AXIAL MOMENTUM,LBF
DSWIRL - SWIRL NUMBER FOR INCREMENT,DIMENSIONLESS
DGP - INCREMENTAL AXIAL PRESSURE FORCE,LBF
DRECIR - RECIRCULATION PARAMETER FOR INCREMENT,PSFA
       - (NEGATIVE MEANS RECIRCULATION ZONE)
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RADIAL AND AXIAL BURNER AIR SWIRL SYSTEM
CONSTANT DENSITY, ADIABATIC, ISENTROPIC
FREE VORTEX IN ANNULAR SPACE, FORCED VORTEX IN SWIRL VANE
LINEAR SWIRL VANE EXIT ANGLE VERSUS RADIUS
PLUME OUTER (TIP) STREAMLINE EQUAL TO EXIT PRESSURE
VERSION 1.54  4/06/89


IPP, INNER SPIN OPEN 90 DEG, SLIDE 5 INCH OPEN, 40 INCH SWIRLER


RADIAL FLOW DAMPER VANE EXIT GEOMETRY
    NUMBER OF VANES=        0
    THICKNESS,FT=    .00000
    SPAN LENGTH,FT=    .41660
    RADIUS,FT=   1.66670
    ANGLE (+C.W.LOOKING UPSTREAM),DEG=    .00000

AXIAL FLOW ANNULAR SPACE AND SWIRL VANE EXIT GEOMETRY
    NUMBER OF VANES=       40
    THICKNESS,FT=    .01042
    ANNULAR SPACE TIP RADIUS,FT=   1.66670
    ANNULAR SPACE HUB AND VANE TIP (SPLITTER) RADIUS,FT=   1.66660
    VANE HUB RADIUS,FT=    .91660
    VANE ANGLE (+C.W.LOOKING UPSTREAM) VS.SPAN (1.0 IS SPLITTER), (DEG/DECIMAL)
    ( 64.00/1.0000)    ( 40.00/ .1111)    (   .00/ .0000)    (


DELTA P EQUALS 1.99 INCH H2O (OUTER REGISTER OPEN 40 DEG)


BURNER SWIRL SYSTEM CONDITIONS
    INLET TEMPERATURE,DEG F= 650.00000
    INLET PRESSURE,PSFA=2126.58000
    EXIT PRESSURE,PSFA=2114.70000
    VANE CASCADE DYNAMIC HEAD LOSS,"q"=   1.00000
    ANNULAR SPACE DYNAMIC HEAD LOSS,"q"=   1.00000


RADIAL DAMPER VANE EXIT CONDITIONS


R=   1.66670  PT=2126.58000  PS=2121.45800  TT=1109.70000  TS=1108.93600
V=  95.79740  VR=  95.79740  VT=    .00000  AT=    .00000  THROAT=   4.36271

FLOW,LBM/SEC=  14.99981
CIRCULATION,SQ FT/SEC=    .00000


AXIAL ANNULAR SPACE EXIT CONDITIONS


STREAMLINE R= 1.6667 PT=2121.5 PS=2114.7 TT=1109.7 TS=1108.7 RHO= .0360
  V= 110.2 VZ= 110.2 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

STREAMTUBE R= 1.6666 DM=  .0041 DGT=  .000E+00 DGX=  .142E-01 DSWIRL=   .000
                               DGP=  .000E+00 DRECIR=   13.6  (12.1)

STREAMLINE R= 1.6666 PT=2121.5 PS=2114.7 TT=1109.7 TS=1108.7 RHO= .0360
  V= 110.2 VZ= 110.2 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0000

FLOW,LBM/SEC=    .00415
SWIRL NUMBER=  .000  REFERRED TO RADIUS,FT= 1.66670
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2114.7 (2116.22)


AXIAL SWIRL VANE EXIT CONDITIONS


STREAMLINE R= 1.6666 PT=2121.5 PS=2114.7 TT=1109.7 TS=1108.7 RHO= .0360
  V= 110.2 VZ=  48.3 VT=  99.0  A= 64.00  B= 64.00  P=   .00 TAU= .0908

STREAMTUBE R= 1.6255 DM= 1.4308 DGT=  .712E+01 DGX=  .229E+01 DSWIRL=  1.866
                               DGP= -.114E+00 DRECIR=    2.6  (1.08

STREAMLINE R= 1.5833 PT=2121.5 PS=2114.4 TT=1109.7 TS=1108.6 RHO= .0360
  V= 112.4 VZ=  54.5 VT=  98.3  A= 61.00  B= 61.00  P=   .00 TAU= .0864

STREAMTUBE R= 1.5422 DM= 1.5280 DGT=  .715E+01 DGX=  .274E+01 DSWIRL=  1.566
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                                 DGP= -.333E+00 DRECIR=    3.0 (1.48)

STREAMLINE R= 1.4999 PT=2121.5 PS=2114.1 TT=1109.7 TS=1108.6 RHO= .0360
  V= 114.6 VZ=  60.8 VT=  97.2  A= 58.00  B= 58.00  P=   .00 TAU= .0835

STREAMTUBE R= 1.4589 DM= 1.6074 DGT=  .703E+01 DGX=  .320E+01 DSWIRL=  1.319
                                 DGP= -.535E+00 DRECIR=    3.5 (1.98)

STREAMLINE R= 1.4166 PT=2121.5 PS=2113.8 TT=1109.7 TS=1108.6 RHO= .0360
  V= 116.9 VZ=  67.1 VT=  95.8  A= 55.00  B= 55.00  P=   .00 TAU= .0816

STREAMTUBE R= 1.3756 DM= 1.6679 DGT=  .676E+01 DGX=  .364E+01 DSWIRL=  1.113
                                 DGP= -.718E+00 DRECIR=    4.1 (2.58)

STREAMLINE R= 1.3333 PT=2121.5 PS=2113.5 TT=1109.7 TS=1108.5 RHO= .0360
  V= 119.2 VZ=  73.4 VT=  93.9  A= 52.00  B= 52.00  P=   .00 TAU= .0808

STREAMTUBE R= 1.2923 DM= 1.7088 DGT=  .637E+01 DGX=  .407E+01 DSWIRL=   .939
                                 DGP= -.881E+00 DRECIR=    4.7 (3.18)

STREAMLINE R= 1.2499 PT=2121.5 PS=2113.2 TT=1109.7 TS=1108.5 RHO= .0360
  V= 121.5 VZ=  79.7 VT=  91.7  A= 49.00  B= 49.00  P=   .00 TAU= .0809

STREAMTUBE R= 1.2090 DM= 1.7293 DGT=  .587E+01 DGX=  .445E+01 DSWIRL=   .791
                                 DGP= -.102E+01 DRECIR=    5.4 (3.88)

STREAMLINE R= 1.1666 PT=2121.5 PS=2112.9 TT=1109.7 TS=1108.4 RHO= .0360
  V= 123.8 VZ=  86.0 VT=  89.1  A= 46.00  B= 46.00  P=   .00 TAU= .0819

STREAMTUBE R= 1.1257 DM= 1.7288 DGT=  .529E+01 DGX=  .479E+01 DSWIRL=   .664
                                 DGP= -.113E+01 DRECIR=    6.2 (4.68)

STREAMLINE R= 1.0833 PT=2121.5 PS=2112.6 TT=1109.7 TS=1108.4 RHO= .0360
  V= 126.1 VZ=  92.2 VT=  86.0  A= 43.00  B= 43.00  P=   .00 TAU= .0837

STREAMTUBE R= 1.0424 DM= 1.7069 DGT=  .466E+01 DGX=  .505E+01 DSWIRL=   .554
                                 DGP= -.122E+01 DRECIR=    7.0 (5.48)

STREAMLINE R=  .9999 PT=2121.5 PS=2112.3 TT=1109.7 TS=1108.3 RHO= .0360
  V= 128.3 VZ=  98.3 VT=  82.5  A= 40.00  B= 40.00  P=   .00 TAU= .0866

STREAMTUBE R=  .9592 DM= 1.8901 DGT=  .217E+01 DGX=  .682E+01 DSWIRL=   .191
                                 DGP= -.128E+01 DRECIR=   11.0 (9.48)

STREAMLINE R=  .9166 PT=2121.5 PS=2112.0 TT=1109.7 TS=1108.3 RHO= .0360
  V= 130.5 VZ= 130.5 VT=    .0  A=   .00  B=   .00  P=   .00 TAU= .0724

FLOW,LBM/SEC=  14.99800
SWIRL NUMBER=  .849  REFERRED TO RADIUS,FT= 1.66660
RECIRCULATION PARAMETER REFERRED TO PRESSURE,PSFA=  2114.7 (2116.22)


CONVERGED, NUMBER OF FLOW ITERATIONS=          6

SWIRLER TO TOTAL FLOW RATIO=     1.000


SYMBOL DEFINITION FOR EXIT CONDITION TABLES
    R - RADIUS,FT
    PT,PS - TOTAL,STATIC PRESSURE,PSFA
    TT,TS - TOTAL,STATIC TEMPERATURE,DEG R
    V,VZ,VT,VR - TOTAL,AXIAL,TANGENTIAL,RADIAL VELOCITY,FT/SEC
    A - PROJECTED FLOW ANGLE (ON PLANE PERP TO RADIAL LINE),DEG
    B - FLOW ANGLE (MERIDIONAL PLANE),DEG
    P - STREAMLINE SLOPE,DEG
    THROAT -AREA,SQ FT
    AT - THROAT ANGLE (OFF RADIAL LINE IN PLANE PERP TO C-L),DEG
    RHO - DENSITY,LBM/CU FT
    TAU - BLOCKAGE,FRACTION


SYMBOL DEFINITION FOR INCREMENTAL VALUES
    DM - INCREMENTAL MASS FLOW,LBM/SEC
    DGT - INCREMENTAL TANGENTIAL MOMENTUM,FT LBF
    DGX - INCREMENTAL AXIAL MOMENTUM,LBF
    DSWIRL - SWIRL NUMBER FOR INCREMENT,DIMENSIONLESS
    DGP - INCREMENTAL AXIAL PRESSURE FORCE,LBF
    DRECIR - RECIRCULATION PARAMETER FOR INCREMENT,PSFA
           - (NEGATIVE MEANS RECIRCULATION ZONE)
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In this paper Cleveland Elect. used impellers to stabilize flame. I'm surprised RJM did not know of this application

CODE 03014

BURNER IMPROVEMENT PROGRAM ON A 650 MW SUPERCRITICAL STEAM GENERATOR - A CASE STUDY

Richard C. Harrington - Cleveland Electric Illuminating Company
Robert P. Kaltenbach - Burns & McDonnell
Harold L. Stratton - Burns & McDonnell

ABSTRACT

This paper provides an overview of combustion improvements achieved on a 650 MW B&W coal-fired boiler. A number of long-standing fireside problems have been significantly reduced as a result of burner modifications and an in-depth testing and optimization program.

Two of The Cleveland Electric Illuminating Company's largest boilers have suffered from a history of problems due to poor combustion and a small furnace. Availability was adversely affected due to slag accumulation on pendant surfaces and furnace bottom, as well as waterwall wastage and overheating of secondary superheater tubing.

CEI embarked upon a program to improve combustion in Avon Lake Unit #9 by replacing the original "daisy chain" burner air registers with a new air register. The burner modifications were made in the fall of 1985 at the same time the boiler was converted to balanced draft. After start-up, significant firing difficulties became apparent. The fire was high in the furnace and, at the same time, burner components were overheating.

An extensive test program was initiated that quantified the problems and permitted developing corrective actions. The indicated corrective action was implemented and final performance verification tests were conducted. The testing has shown excellent results.

BURNER IMPROVEMENT PROGRAM ON A 650 MW SUPERCRITICAL STEAM GENERATOR - A CASE STUDY

INTRODUCTION

The Cleveland Electric Illuminating Company's two largest non-nuclear generating units, Avon Lake Unit #9 and Eastlake Unit #5, have suffered from a long history of combustion related problems. Problems directly attributed to coal quality and the overall combustion process have cost millions of dollars in availability losses and forced outages on both units. It has been conservatively calculated that availability losses on Avon #9 due to combustion and slagging problems amounted to 3-1/2% during the period from 1976 to 1985. During the years 1984 and 1985, Eastlake #5 suffered availability losses of 5% just due to forced outages alone. The problems in every instance can be related to one or more of the following factors: poor coal quality, marginal furnace design and poor burner performance.

In 1985, CEI embarked on a program to improve the burner performance of these two 650 MW B&W supercritical boilers. Much of the blame for poor combustion was attributed to the failure of the burner secondary air register to achieve efficient mixing of the coal and air. The program began with replacement of the original B&W circular, "daisy chain" air registers in Avon #9 with a completely new air register concept. A complication was introduced because at the same time of the air register change out the boiler was also converted from pressurized to balanced draft operation.

The subsequent performance of the boiler with respect to combustion conditions after the conversion was very disappointing. Slagging of the upper furnace pendant tube surfaces worsened rather than showed improvement. Additionally, it soon became apparent that significant thermal and mechanical damage was being imposed upon various burner components. Combustion consulting expertise and testing management expertise was hired from Burns & McDonnell to diagnose the problems and recommend corrective action.

The Avon #9 testing program that followed evolved into an extensive R&D program. A high level engineering team, formed with Burns & McDonnell and CEI personnel, has worked extensively on this project since March 1986 and is still in effect. The outcome of this effort has been highly beneficial and many unique and innovative studies yielded valuable design information. Part of what was learned from these studies is presented in this paper. As a result of applying this information to make burner refinements and operational changes, a dramatic improvement in combustion has been achieved. This has resulted in a very good superheater pendant slagging condition, lower furnace exit gas temperature (FEGT), reduced ash accumulation in the back pass horizontal tube banks and the ability to burn a previously badly slagging coal with much fewer problems. Other parameters such as economizer exit temperatures, unburned carbon, CO levels and NOX levels are as good or better than expected. As a result of the success on Avon #9, CEI has decided to go ahead with the same improvements to the burners on the Eastlake #5 boiler during the balanced draft conversion this fall.

Avon #9 still experiences problems in the lower furnace due to wet runny slag and burner eyebrows. Work is continuing on this problem and it is felt that recent testing concluded this year has identified the flame mechanism responsible for this condition and the modification needed to solve it. This same work has provided some insights into the cause of high CO levels below the burners. There is reason to believe that the modification expected to improve lower furnace slagging could also reduce transient conditions which promote localized high CO levels.

Combustion Problems

Both Avon #9 and Eastlake #5 boilers are B&W 650 MW supercritical (once through) UP type boilers. Avon #9 went into service in 1970 and Eastlake #5 in 1972. These boilers were designed to be pressurized furnaces and were equipped with gas recirculation systems in the upper furnace for flue gas tempering. Both boilers burn high volatile bituminous Ohio coals from a variety of mines. Typical ash contents of these coals range from 9%-14% with ID (initial deformation) temperatures between a low of 1,910°F to a normal around 2,000°F. Fig. No. 1 is a cross-sectional view of the Avon #9 boiler (Eastlake #5 is very similar) and Table No. 1 provides typical coal analysis values for the two boilers.

Industry experience has borne out that boilers of this design and vintage were severely under size in furnace volume and burner zone heat release area. Also, convection pass horizontal tube spacing was too close resulting in flue gas velocities of much higher values than acceptable by current practice. As a result, both boilers have suffered high rates of tube errosion in the convection pass. In the case of CEI's boilers, the small furnace design was further aggravated when the gas recirculation systems were abandoned in 1981 and 1982. This action was necessitated by intolerable living conditions within the plant and the high maintenance required by these systems. The convection pass tube errosion problem has been worsened by high localized gas velocities created by piles of fly ash that accumulate on the horizontal tube banks. The ash accumulation occurs as a result of ash chunks that are blown off of the superheater pendants and lodge between the tubes, blocking the free space. This is caused by heavy slagging of the pendant superheater and frequent sootblowing.

In spite of careful operator attention, occasionally the superheater slagging condition can get beyond the operator's control. With a pressurized furnace, the size and rate of growth of the "clinker" forming on the pendants can be difficult to judge. In 1984 and 1985, Eastlake #5 experienced three large clinker falls that resulted in extensive damage to the furnace bottom slope. The frequent formation of superheater clinkers has been a constant concern at this plant. The locations where clinkers and fly ash piles frequently form are noted on Fig. No. 1.

Poor combustion has also been responsible for a long history of reducing (atmosphere) corrosion of lower furnace waterwalls. Waterwall tube wastage, particularly on the lower sidewall panels, has required wholesale replacement of these panels in both furnaces. Various attempts to remedy this problem have been unsuccessful. As a result, present practice is to rely heavily upon aluminizing for protection.

High furnace exit gas temperature (FEGT) has been responsible for a raft of maintenance problems at both plants. These include pendant and platten warpage, accelerated aging of tube metal and the promotion of liquid phase (slag) corrosion. Prior to the testing program, it was believed that not much could be done to reduce FEGT without adding additional heat absorption surface in the furnace. However, since then results from the combustion testing have shown that FEGT is influenced by flame condition and can be lowered.

In a study to correct these problems, B&W recommended extensive furnace modifications which included spreading out the burners, converting the windbox to a compartmentized windbox, relocating horizontal superheater surface into the furnace as wing walls to lower FEGT and opening up convection pass tube spacing. However, the capital cost of $37 million for each boiler made this solution economically prohibited. As an alternative, CEI decided to work on improving the boiler combustion process. The scope of the resulting project included new burner air registers, windbox corrections and a boiler diagnostic system.

Burner Modifications

Both the boilers have 48 burners, opposed fire, front and rear. Secondary air for combustion is supplied via a wrap-around (plenum principle) windbox which provides air to all the air registers. Each unit has six pulverizers which provide coal and primary air to eight burners.

The original burner air registers were the B&W cell burner, commonly referred to as the "daisy chain" air register. The coal nozzle employed the B&W conical ring type impeller to function as a coal spreader. The arrangement of the original "daisy chain" burner assembly is illustrated in Fig. No. 2.

Previous attempts by CEI to improve burner performance were frustrated by the inability to keep the "daisy chain" air register operable. This air register utilizes a series of linkages located around the periphery of the air register to drive the multiple circumferential damper blades open or closed. Binding of this linkage, making it impossible to move the damper blades, has been a long-standing problem. This is a common complaint with this air register. Consequently, CEI, as in the case of most utilities, had to permanently fix the air registers in the open position. This seriously hampers the ability to efficiently mix fuel and air to achieve optimum combustion at any condition other than when all burners are in service. When operating the boiler with one or more pulverizers off, the air registers of idle burners rob air from the active burners. This can result in incomplete combustion creating localized reducing atmospheres and delayed burning of the coal.

In the fall of 1985, after a period of investigation and testing, CEI elected to install in the Avon #9 boiler the air registers illustrated in Fig No. 3 and also shown in the photos in Fig. No. 4. This is a highly turbulent burner with stable flame characteristics. These were set for three-position air inlet damper control-open, light-off and closed. No changes were made to the coal nozzle or the B&W conical ring impeller. Later, after it was determined that the conical ring type impeller is incompatible with this air register, the impeller was changed to the turbine blade design shown in the photo.

Part of the program included an investigation of the wrap-around windbox for possible airflow maldistribution. For many years, the plant operations people had been convinced that a major portion of airflow went to the front of the windbox, leaving rear burners starved. Extensive model testing could not uncover any real problems with the windbox design; however, to improve the turning of air to the rear windbox, baffles were installed behind the air foils in the right and left side supply ducts. It isn't possible to include details of this investigation here, except that it was confirmed by airflow testing of the boiler prior to start-up (after installing the new air registers) a good flow balance was achieved between all air registers. This work again confirmed the need for good working air registers.

Testing Program

Soon after the boiler was back in operation, several problems of serious concern developed. These included an increase in tendency toward slagging the superheater pendants and thermal damage to the conical ring impellers. Furthermore, an inspection of the furnace during a forced outage four months after start-up revealed bending and overheating of the 304 SS air register vanes, overheating of several coal nozzles and warping of the superheater pendants. The photos shown in Fig. No. 5 are of damaged burners after eight months of service.

The problems experienced were unexpected since a considerable amount of testing had preceded the installation with very satisfactory performance. The initial tendency was to blame the balanced draft conversion for being somehow responsible. In order to determine the cause of the problems, Burns & McDonnell was hired to establish an investigative testing program.

The testing program evolved from an initial series of investigative tests on the Avon #9 boiler to include burner airflow characterization testing at Nels, Incorporated, testing of prototype modifications in a 50 MW B&W front-fired boiler at the Ashtabula Plant and final post-modification testing of the Avon #9 boiler following the fall 1986 burner modifications. A time line depicting the 1986 Avon #9 testing program is provided by Fig. No. 7. More recently, preconversion base line testing has been conducted on Eastlake #5 and testing of several new coal pipe impeller configurations was completed on the 50 MW boiler.

Results of Investigation

The initial problem investigation included a series of tests that concentrated primarily on varying excess O_2. Individual pulverizer-burner operation was carefully scrutinized at this time for correct air/fuel ratio and possible fuel line flow imbalance. In order to evaluate the thermal damage to burner components, air register vanes, coal nozzle and burner impellers were equipped with thermocouples. To better assess the state of combustion in the upper furnace, HVT traverses and gas analyses were taken at the face of the superheater pendants. To provide data on economizer exit conditions, traverses were performed to obtain velocity and temperature profiles and CO, O_2 and NO_x levels. Coal and fly ash analyses were also obtained on a routine basis.

All tests were conducted at a 620 MW condition as this provided sufficient load where problems were readily apparent yet was not so severe as to prevent an eight to ten-hour test observation. During this time, sootblowing was minimized. A very important part of the evaluation of each test depended upon visual furnace observations for characterizing the fire and superheater pendant slag condition. This evaluation was performed exclusively by one skilled individual throughout the program.

The early investigation was intended to uncover the causes of problems, recommend corrective action and provide interim operating recommendations for improving daily operation. The principle problems being experienced by the plant included:

- Heavy superheater slagging accompanied by high FEGT and suspected superheater pendant warpage.

- Excessive impeller wastage and loss of impellers due to burnout of the inner cone.

- Overheating and warping of coal nozzles accompanied by mechanical and thermal damage to the air register vanes.

The testing confirmed these problems to be severe and chronic. The HVT data revealed very high FEGT's, typically around 2,370°F, with high peak temperatures in the center of 2,600°F. The gas analysis taken in this region revealed average CO levels frequently around 1,000 ppm, indicating significant combustion was still taking place high in the furnace. When burners were taken out of service, particularly the upper center burners which are in the hottest zone, coal nozzle temperatures and impeller temperatures shot up to 1,960°F and remained there despite various efforts to provide cooling. This far exceeded the maximum design temperature of 1,800°F for 309 SS, of which these components are made. It also made it nearly impossible to achieve sufficient cooling upon start-up of a pulverizer to avoid the potential of coking the impeller and nozzle.

The testing of the Avon #9 boiler provided some answers, but raised even more questions. Nevertheless, several conclusions were possible. These were: 1) there was an apparent large in-leakage of air, as evidenced by a frequent furnace condition that appeared to be very tight on air while at the same time excess O_2 levels at the economizer exit were more than adequate; 2) to compensate for the air in-leakage and to avoid slagging the superheater, it was necessary to operate at a higher level of excess O_2 than originally recommended by B&W; and 3) making a concerted effort to hold the pulverizer air/fuel ratio down as much as possible provided some improvement.

A forced outage gave an opportunity to perform a closer inspection of the burner and coal nozzle damage, which allowed one additional conclusion - most of the air register vane damage was a result of coal nozzle movement allowing the pipe to bend the vanes.

At this point, the inability to locate a significant boiler casing leak as well as adequately explain the high coal nozzle and impeller temperatures was of sufficient concern that it was decided to move into a laboratory testing program. This consisted of a two phase series of tests, run concurrently, which included: airflow performance testing and flow characterization of full size air registers; and testing of modifications developed in the lab in the 50 MW, six burner, Ashtabula boiler.

With the additional knowledge obtained from the airflow testing, the flame observations and temperature data from the 50 MW boiler, and with the discovery of the Avon #9 air leak, it became possible to nail down the causes of all the problems. Briefly, these are described as follows:

Source of Leak. A major leak was discovered during a furnace inspection which explained earlier suspicions. It is estimated that the leak allowed as much as 24% of the secondary combustion air to bypass the burners and enter the furnace above the burner zone. This was caused by a large area of missing refractory at the tube openings where the second and third pass waterwall mixing zone headers are located. These go around all four walls of the furnace. A view of a partially repaired section of refractory is shown in the photo in Fig. No. 6. It is readily apparent that air passing through these openings travelled directly up the walls with little influence on the combustion process.

Burner Damage. The additional testing showed that some improved cooling of the coal nozzle was achieved by increasing the nozzle to vane clearance from 3/4" to 2". However, through a detailed analysis of Avon #9 damage patterns, it was determined that the leading cause of nozzle and vane overheating was directly related to the nozzle hitting and bending the air register vanes. Increasing the clearance and anchoring the end of the coal nozzle to keep it centered proved to be the solution to the high temperature problems.

Impeller Wastage. An important discovery made from the airflow testing and Ashtabula burner flame observations was the fact that the B&W conical ring impeller is incompatable with the new style air register. The additional testing and Avon #9 impeller temperature data prove conclusively that this impeller in this application is highly subject to coking and blinding. This discovery prompted an investigation of alternative impeller designs which concluded with the adoption of the turbine blade style impeller. This is discussed in more detail further on in this paper.

Impeller Cooling. It was determined that in order to promote cooling of out of service impellers, the previously abandoned practice of retracting impellers should be reinstituted. For this purpose, a new, more reliable retractor design was needed.

Lower Fire in Furnace. A second important discovery from the Ashtabula boiler burner testing was that the turbine blade impeller significantly lowers the fire mass in the furnace. This effect is created when the impeller and air register spin directions are counter to each other. This combination created a very low, dense mass of fire in the lower furnace with no flame tails at the superheater screen tubes whatsoever. Heat absorption into the waterwalls was noticeably increased and absorption at the superheater reduced.

Results of Burner Modifications

During the 1986 fall maintenance outage of Avon #9, the following corrections were made:

- The vane to coal nozzle clearance was increased to 2" and a centering ring was added to prevent the coal nozzle from moving.

- A newly designed impeller retract system was installed.

- The B&W conical ring impellers were replaced with the turbine blade type. These were installed so that the impeller spin rotation opposed the spin of the air register vanes.

- The refractory was replaced at the waterwall mixing zones to eliminate the air leak.

These repairs and modifications created a remarkable improvement in furnace conditions which were immediately obvious. The fire in the furnace was much lower and totally void of raw flame tails. Superheater slagging was no longer a problem, even at maximum load of 660 MW, or with the previously "bad" (slagging) coal. Furnace exit gas temperature was lower and more uniform across the furnace. Out of service and in-service burner component temperatures were lower and within acceptable limits. The impeller coking, blinding and wastage problem was completely eliminated. One additional benefit was that ash accumulations on the convection pass horizontal tube banks were noticeably less.

Despite the positive effect the burner changes had on the boiler, it also had a negative impact in the area of wall slagging in the lower furnace. As a result of higher temperatures in the lower furnace, possibly combined with the effect of the broader flame pattern with the new impeller, the boiler began to experience frequent occurrences of wet runny slag on the lower side walls. This slag would run down the walls and onto the bottom slope, where it was necessary to use water lances to remove it. Secondary, the formation of sizable burner eyebrows became more prevalent. The photo in Fig. No. 8 shows a large eyebrow formation.

The post-modification testing performed shortly after the burner changes were completed, demonstrated the need for further work. Additional studies of flame patterns created by different impeller arrangements were recently concluded this year on the 50 MW Ashtabula boiler. The aim of this work has been to investigate the impeller-air register interaction in an attempt to improve flame shape. From these tests, it is felt that modification of the flame shape by impeller changes will reduce the lower furnace temperatures and eyebrow formation. The following sections discuss some of the results from these studies.

Effect on FEGT (Furnace Exit Gas Temperature)

Figure No. 9 is a plot of average FEGT values measured at four locations just in front of the superheater pendants. The data is of Avon #9 before and after the burner modifications. Eastlake #5 boiler baseline test data is also plotted for comparison. This is a pressurized furnace with B&W air registers that are fixed in the open position.

FEGT measurements were made by both the HVT (high velocity thermocouple) method and by a hand-held infrared pyrometer. Since a good correlation was obtained between the infrared readings and the HVT readings (corrected to MHVT), and because of the considerable difficulty and cost involved with performing HVT traverses, the bulk of the FEGT data is based upon corrected infrared measurements. This same method of measurement was also employed to obtain furnace gas temperatures at other locations.

The FEGT values shown on Figure No. 9, therefore, while not precise, are sufficiently accurate for making trends and comparisons. From the plots, it is clear that changing the burner impeller to the turbine blade style and fixing the furnace air leak lowered the FEGT approximately 100°F. Although not proven, it is also strongly suspected that high peak temperatures, which were in excess of 2,600°F, have been reduced even more. When comparing the Avon #9 "after modifications" to the Eastlake #5 data, it is readily seen that Avon #9 has a lower FEGT even though it is balanced draft. The effect of this was observed during the testing of the two units. The tendency for superheater slagging of Eastlake #5 being considerably worse than Avon #9. In comparing the overall upper furnace conditions of the two boilers, Avon #9 with the new burners is much superior.

Effect on Lower Furnace Temperatures

The effect of the impeller modifications on lower furnace temperatures is just the reverse of that on FEGT. The impeller creates a hot, dense fire mass in the lower furnace that produces higher gas temperature measurements in the furnace bottom.

Figure No. 10 gives a comparison of temperatures measured in both Avon #9 and Eastlake #5 at the side wall observation ports below the bottom row of burners. The plots indicate the higher temperature measured at each test condition. Of the two boilers, the data shows Avon #9 to be approximately 80°F higher. Where Eastlake #5 does not have a wall slagging problem in the lower furnace area, Avon #9 does, thus the higher temperature is suspected as the leading cause.

Effect on Furnace Bottom CO (Carbon Monoxide)

During an early on-line inspection of Avon #9, a "halo" could be seen off of the inspection doors indicating high levels of CO in the furnace bottom. While this condition was helped during early tuning and testing, it would occasionally return and remained an ongoing concern. Sample taps were installed under the bottom row of burners to permit monitoring CO levels at that location during the fall 1986 outage. This area was selected because of the high CO in the furnace bottom observed visually and because Avon #9 has had some tube wastage in this area. Unfortunately, there was no prior data to compare to either before replacing the air registers or during the early 1986 operation, however, after restart of the unit there was none of the visually high CO evident.

Subsequent baseline testing has been done on the Eastlake #5 with similar CO data collection. This boiler still has the original daisy chain registers, but has not experienced the same unacceptable level of tube wastage under the burners. The Avon #9 and Eastlake #5 CO data is summarized in Table No. 3. When reviewing the CO data it is noted that for burner stoechiometrics down to 0.85 there appears to be little if any effect on the quantity of CO detected under the burners. For this reason and because the Eastlake #5 CO data is based on operating with a pressurized furnace, the differences in the economizer exit O_2 shown in Table No. 3 is thought to have very little effect on the results. The data also shows that generally the two units are experiencing nearly the same levels of CO. A notable exception to this is with all mills in service. In this case, Avon #9 has 723 ppm and Eastlake #5 has 3,873 ppm average. Also, economizer exit CO levels were taken during the first two tests listed in Table No. 3. The average exit CO value during these tests was 11 ppm and 42 ppm, well below the acceptable level of 200-300 ppm.

Information concerning acceptable levels of CO at this location (under the burners) has not been available. However, since Eastlake #5 has not experienced any siginificant tube wastage problem in this area, it is considered that the CO levels on Avon #9 are acceptable but higher than it could be. While there is evidence to support the levels of CO present as being acceptable, the overall objective is to reduce the CO level under the burners to be consistently in the order of 500 ppm or less.

To this end, the CO data was carefully reviewed and it was found that statistically the higher CO levels on Avon #9 were linked to "B" mill and excessive PA flows. B mill, which since was overhauled, had an excessively worn throat ring causing poor operating characteristics. Due to the worn throat ring, B mill was consistently operated at high PA flows to prevent coal from being rejected out the reject hopper. Also, with careful review of the data, it was shown that there were numerous instances when B mill was off that there was higher than design PA flow to one or more mills. Table No. 2 provides a statistical review of this information.

The table shows that the CO level under the burners can be substantially reduced by maintaining the PA flows at or near design. When substantially higher than design PA flows are required for mill operation, the need to repair the mill becomes evident.

Impeller Interaction Studies

The basic problem with the original B&W conical ring impeller was due to a strong reverse flow generated in the central region associated with the new air register which causes the ignition point of the flame to burn within the inner part of the ring type diffuser. This recirculating condition causes the B&W impeller to be prone to blind with ash deposits which results in frequent periods of extremely poor burner performance. It was possible to visually observe blinded impellers in the Ashtabula boiler and see the detrimental effect on the flame.

The mechanism behind this phenomena was first discovered in the laboratory airflow testing conducted on full size air registers. As part of this testing, the wake behind the B&W impeller was measured in both the new air register and the B&W daisy chain register. Figure No. 11 shows the radically different wake profiles with the two air registers. In the new air register, the wake is substantially compressed. The reason for this compression is explained by other testing conducted which looked at the velocity profile across the burner throat opening created by the two registers. This work showed that the new register has a more uniform profile which results in higher velocities in the center area of the burner throat than exhibited by the B&W daisy chain. These higher velocities apparently reduce the ability of the conical ring type impeller to spread the primary air and coal.

The new turbine blade impeller was developed as a measure to eliminate the negative effects of the reverse flow phenomenon which would in turn restore burner performance by maintaining a consistent flame pattern. The new impeller concept was essentially made up as an impromptu solution to solve a problem that was not previously known about before the testing began at Ashtabula. Evaluations were made at the time which indicated that the counterflow relationship between the impeller and the air register provided the best flame pattern. This relates to the primary air and fuel flow coming across the impeller in a direction opposing the secondary air distribution coming from the air register. Both parallel (complementary) and counter (opposed) relationships were evaluated at the time when the new impeller concept was initially applied to the boiler at Ashtabula. The opposed application appeared to produce the best flame due to having a larger flare shaped pattern. Based on the knowledge learned from the Ashtabula testing efforts, the counterflow concept was applied to all of the burners on Avon #9 boiler during the fall burner restoration outage.

Recent Impeller Testing

The basic approach to the recent impeller testing program at Ashtabula was to check the performance of different styles of blade type impellers installed in both applications of counter and parallel flow arrangements. Plus, similar tests were performed using the original B&W conical ring impeller and also without any impeller being used at all. The latter two tests were added to the test sequence to record the potentially negative effects produced by both conditions, relating to the fact that the B&W impeller is not compatible with the air register and the boiler manufacturer recommends that burners be removed from service in the event of an impeller failure.

Specific burner performance was judged solely on visual observations. The results of each case were recorded with the use of VHS video equipment in order to make reliable comparisons after the testing was complete. Numerical data was taken during each test to assure that consistent conditions were maintained during all of the testing. However, final judgments to make flame performance evaluations were essentially made using the knowledge gained from the video tapes. This proved to be a real asset in meeting the objectives of the testing program. The observations from only two of the tests are reported here.

Counterflow Turbine Blade

The fabricated turbine blade type impeller installed in an opposed relationship produces a flame pattern which departs the distribution of the secondary airflow path. This phenomenon is basically caused by the interaction of the opposed secondary air and the primary air spin relationship. In the case of the two test burners, the bulk of the flame pattern is directed in an askewed direction towards the center of the furnace. The secondary air distribution for both of these burners rotates towards the side walls of the furnace, thus the counterflow impellers were installed with their effective rotation being towards the center of the setting.

The counterflow arrangement produces an extremely wide angle flare type pattern which totally fills the burner throat and generally moves straight away from the burner opening. However, the flame pattern proper, when viewed from the rear of the furnace, can be seen to have high velocity spike tails on the outboard side and the majority of flame is traveling towards the center of the furnace. This results in most of the combustion process of a burner taking place beyond the boundary of the path of secondary air coming from that burner and to depend on air being supplied by an adjacent burner to support complete combustion. In reality, the adjacent burner might also be in operation or it might be out of service at any given time. This impeller set-up produced flames that had good ignition, stayed close to the tip of the burner nozzle and looked excellent when viewed from the side of the burner next to the burner throat opening. The negative aspect of the flame travel towards the center of the furnace was obvious when looking towards the front from the rear and by observations made looking down from the upper elevations. The flame pattern generally caused a wide body of fire that filled the lower furnace region before the movement approached the rear wall. The photos shown in Fig. No. 12 were taken from videotapes of these flames.

Parallel Flow Turbine Blade

The fabricated turbine blade type impeller installed in a parallel relationship produces a flame pattern which is confined to the distribution of the secondary airflow path. This feature is basically due to the result of the complementary action caused by the blending affects of the secondary air and the primary air spin relationship. The parallel flow arrangement produces a tight flame pattern which extends towards the rear of the furnace in the shape of an elongated ribbon of fire. This results in the combustion process of a burner taking place within the space confined to the path of secondary air coming from that burner. The flame is projected straight back from the burner throat opening, but it seems to turn upwards before it reaches the surface of the rear wall. This impeller configuration causes very little flame contact with the back wall. This impeller set-up produces flames that have good ignition, stay close to the tip of the burner nozzle and look excellent when viewed from the side of the boiler next to the burner throat opening. When viewed from this location, the surface of the flame pattern shows signs that the combustion products are twisting or turning in a rotational direction towards the side wall of the furnace. This action is the direct result of the flow from the impeller and the secondary air path being the same relationship. The complementary action results in a flame pattern that is rather compact at the point of origin and the departure angle does not cause the burner throat opening to be totally filled with fire. The photos shown in Fig. No. 13 were taken from videotapes of these flames.

Follow-Up Plan

The Avon #9 lower furnace slagging situation is the one major problem remaining. The evidence strongly points to high temperatures in this region and, potentially, the askewed flame pattern created by the counterflow impeller as the cause. It was concluded from the studies of the impeller testing that the parallel flow impeller produced a superior flame characteristic and better looking furnace condition. In August 1987, half of the burners in Avon #9 will be changed to the parallel flow configuration and the results analyzed.

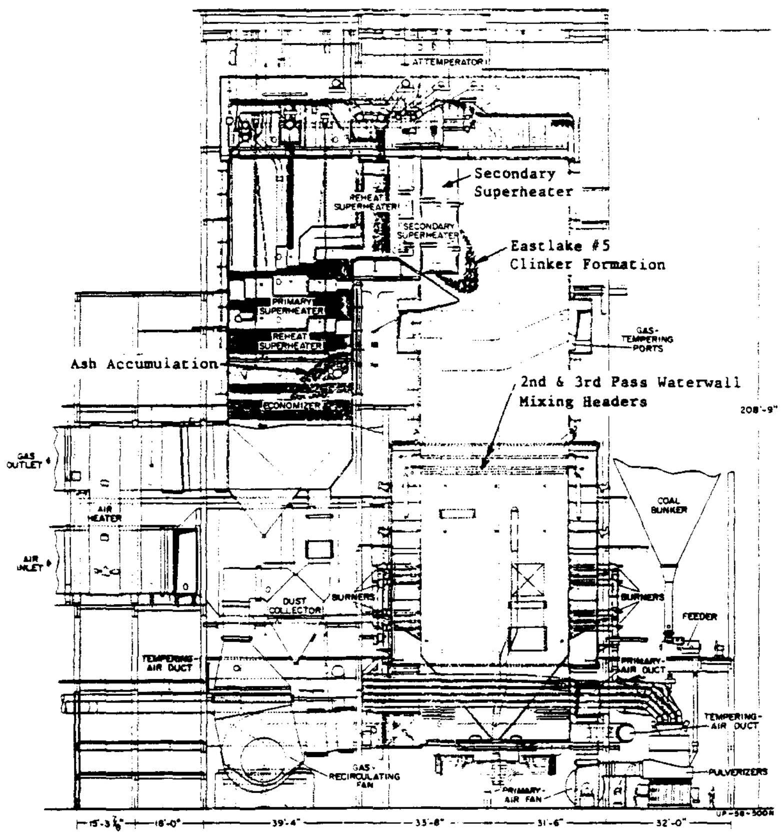

Figure 1. Avon Lake Plant, Unit No. 9, 650 MW, B&W universal pressure boiler clinker and fly ash accumulation are noted.

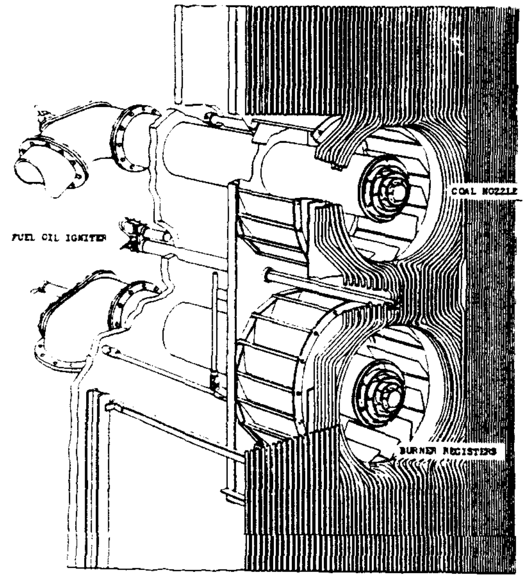

Figure 2. Original B&W cell burner arrangement with the daisy chain air register.

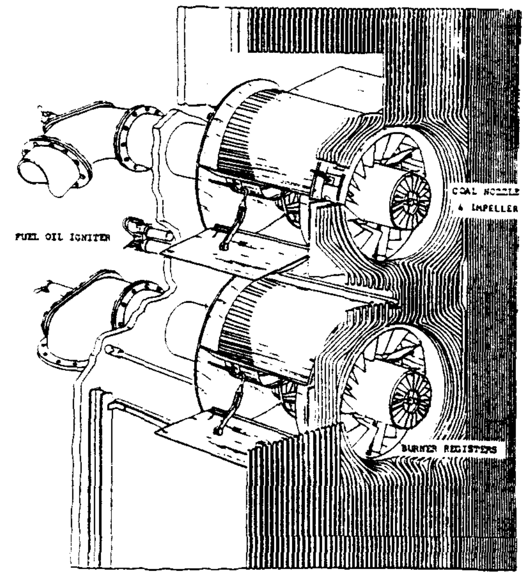

Figure 3. Burner arrangement showing new air registers and turbine blade style impeller.

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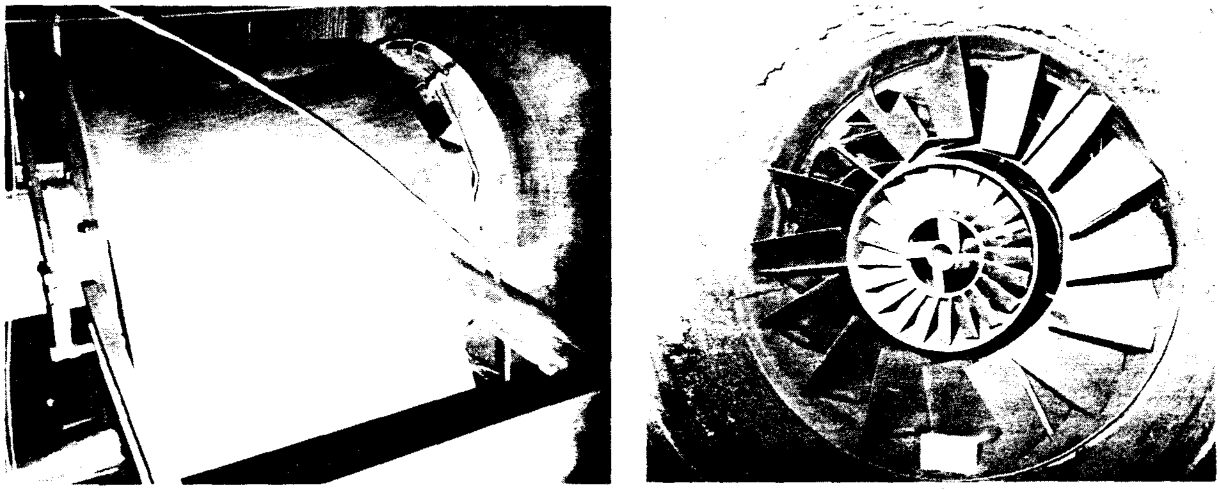

Photo from inside windbox, furnace waterwall on right.

Photo from furnace side. Note turbine blade style impeller.

Figure 4. Photos of new burner air register.

Figure 5. Photos of damaged air registers after initial 8 mos. operation. Note bent vanes due to coal nozzle movement and B&W conical impeller with damaged inner ring.

Figure 6. Missing refractory at 2nd and 3rd pass waterwall mixing zone headers was responsible for air leakage above the burner zone of as much as 24% of the secondary air. White's new refractory being installed.

TESTING PROGRAM										
1986	MAR	APR	MAY	JUNE	JULY	AUG	SEP	OCT	NOV	DEC
INITIAL BURNER TEST	26 TESTS									
NELS AIR FLOW MODEL TEST					49 TESTS					
ASHTABULA BURNER TESTS						6 TESTS				
OUTAGE - AVON #9								OUTAGE		
BURNER PERFORMANCE TEST									6	TESTS
BURNER PERFORMANCE TEST									24	TESTS

Figure 7. Testing performed on Avon No. 9 in 1986 is shown above. This was followed in 1987 with additional impeller tests on the Ashtabula boiler which resulted in impeller changes and follow-up tests on Avon No. 9.

Figure 8. A negative effect of the improvements made during the 1986 fall outage is wet, runny slag on the lower furnace sidewalls and eyebrow formations around burners.

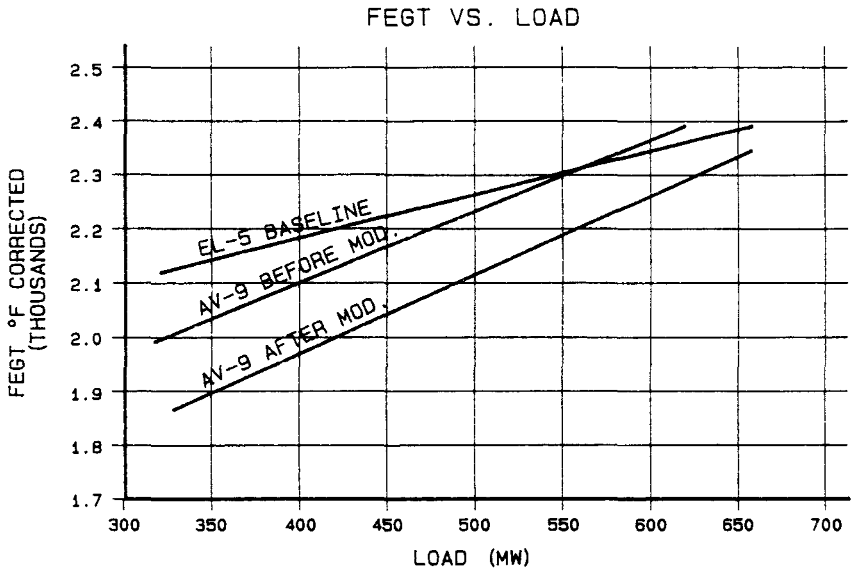

Figure 9. Comparison of furnace exit gas temperature before and after improvements made during the 1986 fall outage. The Eastlake No. 5 baseline data was taken in 1987 and shows the poor performance of the old B&W daisy chain register.

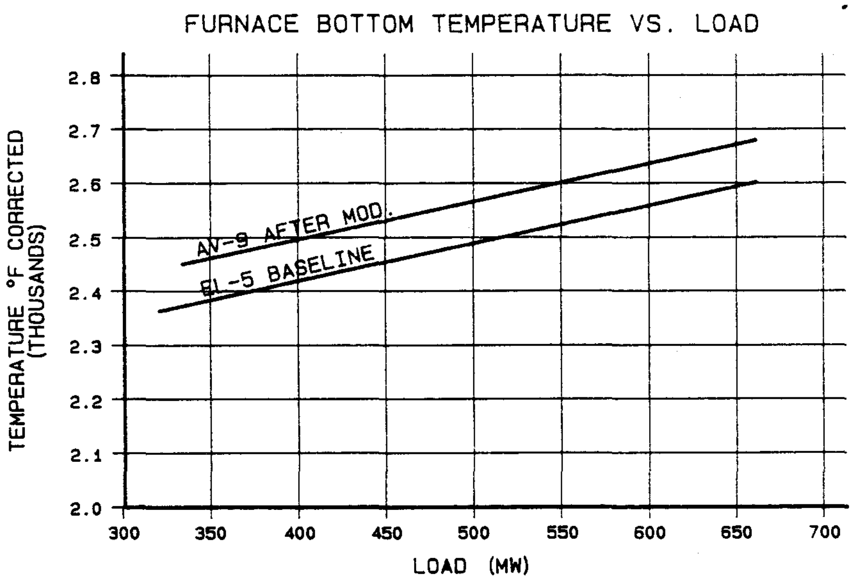

Figure 10. The counter flow turbine impeller creates a dense fire in the lower furnace. This is reflected in higher temperatures below the burners as compared to Eastlake No. 5.

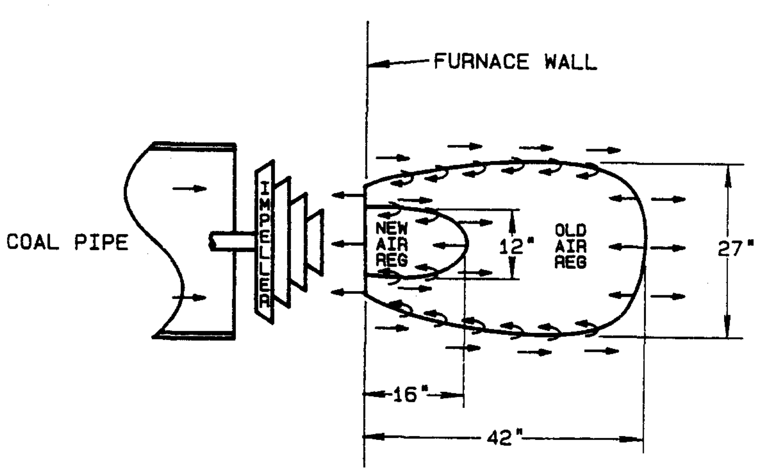

Figure 11. Measured profile of the wake (reverse flow) of the B&W conical ring impeller in the new air register as compared to the original B&W daisy chain.

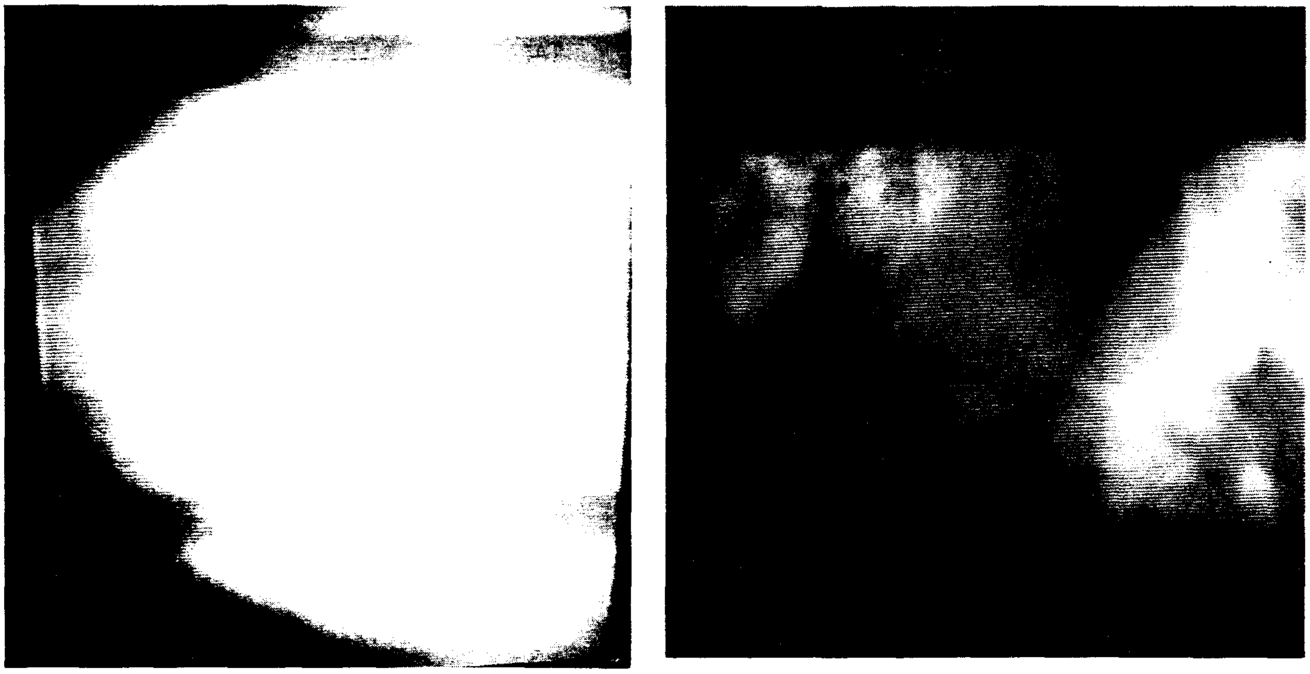

Figure 12. Photos of video taped flames of the turbine blade impeller, counter flow. Side profile (left) shows a broad flare and travels straight away. End view (right) shows the tendency of flames to turn in towards the center (askewed) and occupying the space of the third center burner (which is off).

Figure 13. Similar photos of the turbine blade impeller in parallel flow. Side view (left) shows less flare and a distinct "cork screw" rotation. End view (right) shows much narrower flames that travel straight back. There is plenty of room left for the center burner.

TABLE 1
ANALYSIS AND ASH FUSIONS
TEMPERATURES - TYPICAL COALS

	AV-9	EL-5	
BTU	12,630	12,440	11,740
Ultimate			
C	70.13	69.09	64.64
H	4.89	4.81	4.44
O_2	5.6	4.11	5.72
N_2	1.53	1.10	1.28
Ash	9.21	11.95	14.26
S	2.54	3.58	3.68
H_2O	6.10	5.36	5.98
Prox.			
VM	37.61		
FC	47.08		
Ash	9.21		
H_2O	6.10		
Ash Fusion Temps.			
IT	1947		1913
ST	2035		1974
HT	2124		2002
FT	2347		2054

TABLE 2 - CO LEVELS UNDER BURNERS
NORMALIZED PERCENTAGES

CO (ppm) Range	B Mill On	B Mill Off
Over 1000	50.0%	10%
500 - 1000	22.4%	5%
0 - 500	27.6%	85%
TOTAL	100.9%	100.0%
	PA Flows High	PA Flows Low
Over 1000	45.5%	7.2%
500 - 1000	17.4%	14.2%
0 - 500	37.0%	78.6%
TOTAL	100.0%	100.0%

TABLE 3 - CO DATA

Unit	No of Mills	Load MW	Avg. 02 Econ. Exit	Impeller Type	CO (PPM) Under Burners High	Low	Avg.
AV9	3	316	5.95	Conical	19790	130	4127
	(Various)	404	6.30	Conical	>20000	90	>4955
		352	5.55	T.B.	>20000	60	>3355
		459	6.35	T.B.	19999	330	8787
EL5	3	330	4.70	Concial	14543	228	3511
	(Various)	357	4.50	Conical	17862	108	4204
		411	3.95	Conical	>20000	260	>6587
AV9	4	353	5.65	T.B.	11750	195	5857
	(Various)	552	4.90	T.B.	>20000	175	>11215
EL5	4	331	4.35	Conical	9110	375	4933
	(Various)	406	3.45	Conical	458	128	280
		530	3.65	Conical	662	182	369
AV9	5	450	5.15	T.B.	1100	110	310
	(Various)	451	5.55	T.B.	16160	600	5709
		620	4.05	T.B.	>20000	160	>6707
		620	4.25	T.B.	11385	130	3495
		639	5.25	T.B.	14480	300	6527
EL5	5	390	3.60	Conical	458	85	259
	(Various)	489	3.45	Conical	18891	1433	8792
		649	3.65	Conical	>20000	1289	>11825
AV9	6	654	4.95	T.B.	5995	175	723
EL5	6	503	3.55	Conical	8938	516	3109
		620	3.90	Conical	10915	201	3803
		621	3.70	Conical	10655	159	1882
		656	3.25	Conical	18286	1221	6697

DISCUSSION ON PAPER BY

R. C. HARRINGTON et al

1. R. B. Dooley (EPRI)

   a) Was any thought given to removing the bulk of the impeller and just having a black body?

   b) Was any consideration given to altering the quarl angle? (Although your slides indicate that there wasn't much room to accommodate any quarl angle change).

   R. C. Harrington

   a) Tests and video pictures were made of fires where the impeller was removed entirely. These fires were extreamly poor with a long trashy type fire. The fire is lifted off the end of the coal nozzzle several feet and tends to place a lot of slag on the rear wall of the furnace.

   b) The test burners in the Ashtabula 50 MW boiler have tapered, flared burner throats vs. Av #9 which has a straight cylinderical throat. There is very little difference in the fire shape. However, the straight Av#9 throat does provide a ledge for ash and slag to accumulate.

2. J. A. Arnott (Ontario Hydro, Canada)

   What were the NO_x levels and carbon-in-ash levels before and after the burner conversions.

   R. C. Harrington

   a) NO_x levels - after conversion and modifications values of 0.6#/MMBTU and 0.49 were obtained at full load using EPA method #7. There is some question at this time about the accuracy of this method when more than 1%S coal is burned, NO_x values maybe higher. Good NO_x data prior to the burner work on this boiler is not available. (Tests on EL-5 using EPA method 1E gave 1.83#/MMBTU and 1.27#/MMBTU).

   b) LOI's in the fly ash after conversion but before burner improvements was 1.6%; after improvements this dropped to 0.9%. Eastlake #5, which is pressurized furnace and still equipped with the original B&W burners is 1.8 & 0.7% (two tests).